

REFRIGERATION

and Air Conditioning

JORDAN & FRISTER

Original from
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OF
WISCONSIN

REFRIGERATION

and

Air Conditioning

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To
Freda and Edith



Preface

The purposes of this book are twofold: first, to serve primarily as a college text for senior and graduate study; and second, to serve as a reference for the practicing engineer. The theoretical portions of the text assume that the reader has studied physics and mathematics, and one chapter contains a review of basic thermodynamics. However, several chapters are devoted primarily to refrigeration applications, and to understand them does not require an extensive theoretical background.

As a text, the book is designed for a comprehensive course in refrigeration and air conditioning. It may be adapted to combination courses in refrigeration and air conditioning, in which case portions of the text on advanced refrigeration topics may be omitted if time is limited. It may also be used for courses exclusively in refrigeration, where the student has previously studied the subject of air conditioning. In this case, those portions on air conditioning may be omitted.

The first chapter of the book attempts to instill in the reader an appreciation for the exceedingly interesting history of refrigeration and to arouse his interest in some of the vast potentialities that exist for future development. The broad aspects of the subject having been surveyed, the student is then introduced in the second chapter to the various cycles and processes to be studied analytically later in the text. This is done by means of a series of "sequence" drawings that portray qualitative explanations of the basic refrigeration cycles. Following this, a review of basic thermodynamics is presented in the third chapter. These three chapters may be considered as an introduction to the main portions of the text.

The authors recognize that because of the necessarily broad scope of the subject matter which must be contained in a text on refrigeration, the order of chapter arrangement may not be completely agreeable to all instructors. A specific attempt has been made not to segregate theory, application, and equipment into separate sections. Therefore, descriptive material on equipment follows closely after the presentation of theory applied to such equipment. Specialized advanced topics of refrigeration are placed toward the end of the text, and the chapters on applications are placed at the very end. For those who would prefer to segregate the fundamental theory and applications in that order, the following outline is suggested:

Section I—General:

Chapter 1—The History of Refrigeration

Chapter 2—Basic Refrigeration Cycles and Concepts

Section II—Fundamental Theory:

Chapter 3—Review of Thermodynamics

Chapter 8—Fluid Flow and Heat Transfer

Chapter 9—Psychrometrics

Section III—Refrigeration Theory:

Chapter 4—Thermodynamics of Vapor Refrigeration

Chapter 5—Refrigerants

Chapter 15—Absorption Refrigeration and Chemical Dehumidification

Chapter 6—Air-Cycle, Ejector, and Centrifugal Refrigeration, and the Heat Pump

Chapter 13—Multiple Evaporator and Compressor Systems

Chapter 10—Cooling-Load Calculations

Chapter 16—Low-Temperature Refrigeration

Section IV—Equipment:

Chapter 7—Compression Refrigeration Machines

Chapter 11—Condensers and Evaporators

Chapter 12—Refrigeration Piping

Chapter 14—Refrigeration Control

Section V—Applications:

Chapter 17—Applications of Refrigeration

Chapter 18—Air Conditioning

Chapter 19—Food Preservation

As an aid to both the student and the practicing engineer, extensive documentation by means of footnotes has been provided. In addition, at the end of each chapter a Bibliography has been included, indicating where more complete information may be found. An extensive set of problems, graded in difficulty, has been provided at the end of most chapters. Many of the answers to the problems are provided in the section immediately following the Appendix.

The authors are indebted to Mr. Charles S. Leopold and to his assistants, Mr. Frank Wilder and Mr. James Wolff, for their review and helpful criticism of the entire manuscript; and to Dr. Newman Hall for his suggestions on the chapters on thermodynamics and low-temperature refrigeration. In addition, the authors wish to thank the staff members of the American Society of Refrigerating Engineers and The American Society of Heating and Ventilating Engineers, and the various publishers,

manufacturers, distributors, and other organizations which, as indicated in the text, have granted us permission to reproduce figures and tables. Any errors pointed out or improvements suggested would be greatly appreciated by the authors so that appropriate changes may be incorporated in future revisions.

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PART I
Introduction

CHAPTER 1

The History of Refrigeration

1.1. Refrigeration by Natural Ice. In the early part of 1806 the brig *Favorite* slid into the harbor of St. Pierre, Martinique, with 130 tons of cake ice in her hold. This was probably the first large-scale commercial venture in the refrigeration field, and the owner of this cargo, Frederic Tudor, lost some \$3500 on the venture. Since ice was unknown in Martinique and no storage facilities were available for its preservation, the financial loss would have been much greater but for some quick thinking on the part of the man later destined to become known as the Ice King. Through arrangements with one of the eating-house proprietors he concocted and introduced ice creams in the West Indies, where frozen desserts were virtually unknown.

In later years, by the construction of an icehouse at St. Pierre and by the use of pine sawdust as an insulation during the transportation of his ice cargoes, Tudor turned his idea into an extremely profitable business. He contracted for the cutting of ice in ponds and rivers throughout New England and shipped it throughout the world, not only to the West Indies and to the southern part of our own country but also to such faraway places as South America, Persia, India, and the East Indies. In 1849 his cargoes totaled 150,000 tons of ice; by 1864 he was shipping to 53 ports in various parts of the world. The business he founded changed the lives and habits of people throughout the world, and the methods he used remained essentially the same until they were supplanted in the 1880's by the manufacture of artificial ice.

Tudor's ice empire was the first large-scale venture in refrigeration. Prior to his time ice was used in small quantities in many parts of the world where available without transportation. In fact, refrigeration as an art had been known for thousands of years. In an ancient Chinese collection of poems, Shi Ching, there are references to the use of ice cellars in 1000 B.C. The Greeks and Romans are known to have constructed snow cellars in which they stored tightly compressed snow and insulated it with grass, earth, and manure. Pliny the Elder writes of the ill effects of cooled drinks, and the emperor Nero is said to have invented cooling of beverages by placing their containers in snow. The Indians, Egyptians, and Esthonians chilled water and even produced ice by placing water in shallow, porous clay vessels, then leaving these overnight in holes in the ground. Radiation of heat to the colder interstellar space together

with vaporization of some of the water combined to accomplish the freezing. As early as the fourth century A.D. the East Indians knew that certain salts, such as sodium nitrate, when placed in water would result in a lowering of the temperature.



Fig. 1.1. An early cold-storage refrigeration application—from an 1890 Frick catalog. Courtesy Frick Co.

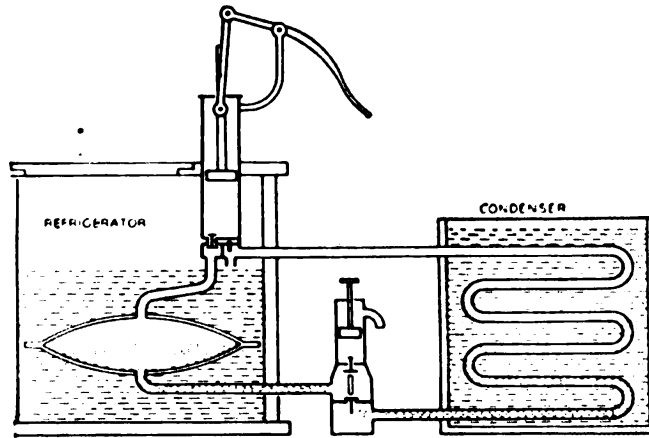
1.2. Artificial Refrigeration in the Nineteenth Century. In the spring of 1863 an anonymous author wrote as follows in the *London Practical Mechanics Journal*:

One hundred years ago the notion of making ice by a machine would have seemed as preposterous as an attempt to call down fire from Heaven; by the popular masses, even the most civilized in Europe, it would have been deemed an impossible but highly impious attempt to usurp or travestie the supposed special powers of Deity alone; but to the best informed, the ice making would have seemed more impractical than even the calling down or diverting the lightnings of heaven. The latter had, in fact, already been done.¹

By this time engineers had designed and built practical artificial refrigeration machines, though their slow-speed, oversized, and even hand-operated equipment appears, in retrospect, somewhat ridiculous.

¹ Quoted in "Ice Making Machine." *Journal of the Franklin Institute*, Vol. 76 (1863), p. 104.

The earliest recorded patent for a refrigeration machine was issued in Great Britain in 1834 to Jacob Perkins, an American. This unit, shown in Fig. 1.2, consisted of a hand-operated compressor, a water-cooled con-



BRITISH PATENT #6,662

to

JACOB PERKINS, GRANTED 1834.

What I claim is an arrangement whereby I am enabled to use volatile fluids for the purpose of producing the cooling or freezing of fluids, and yet at the same time constantly condensing such volatile fluids, and bringing them again and again into operation without waste.

Fig. 1.2. The earliest recorded patent for a refrigeration machine, issued in Great Britain in 1834. From "Some Interesting Refrigeration Inventions," by Robert A. O'Leary. *Refrigerating Engineering*, Vol. 42, No. 5, November, 1941.

denser with a weighted valve at the discharge, and an evaporator contained in a liquid cooler. The unit was designed to be used with ether as the refrigerant.

There were many other nineteenth-century pioneers in refrigeration. In 1851 Dr. John Gorrie of Florida obtained the first American patent for an ice machine, designed to use compressed air as the refrigerant. As a physician he was motivated by a desire to relieve the sufferings of fever patients and others exposed to high temperatures. Professor A. C. Twining of New Haven developed a sulfuric ether machine that preceded Gorrie's; however, he failed to have it patented in the United States until 1853. Dr. James Harrison of Australia also developed a sulfuric ether machine and in 1860 made the world's first installation of refrigerating equipment in a brewery. In 1861 Dr. Alexander Kirk of England con-

structed a cold-air machine similar to Gorrie's; it is said to have consumed one pound of coal for each four pounds of ice produced.

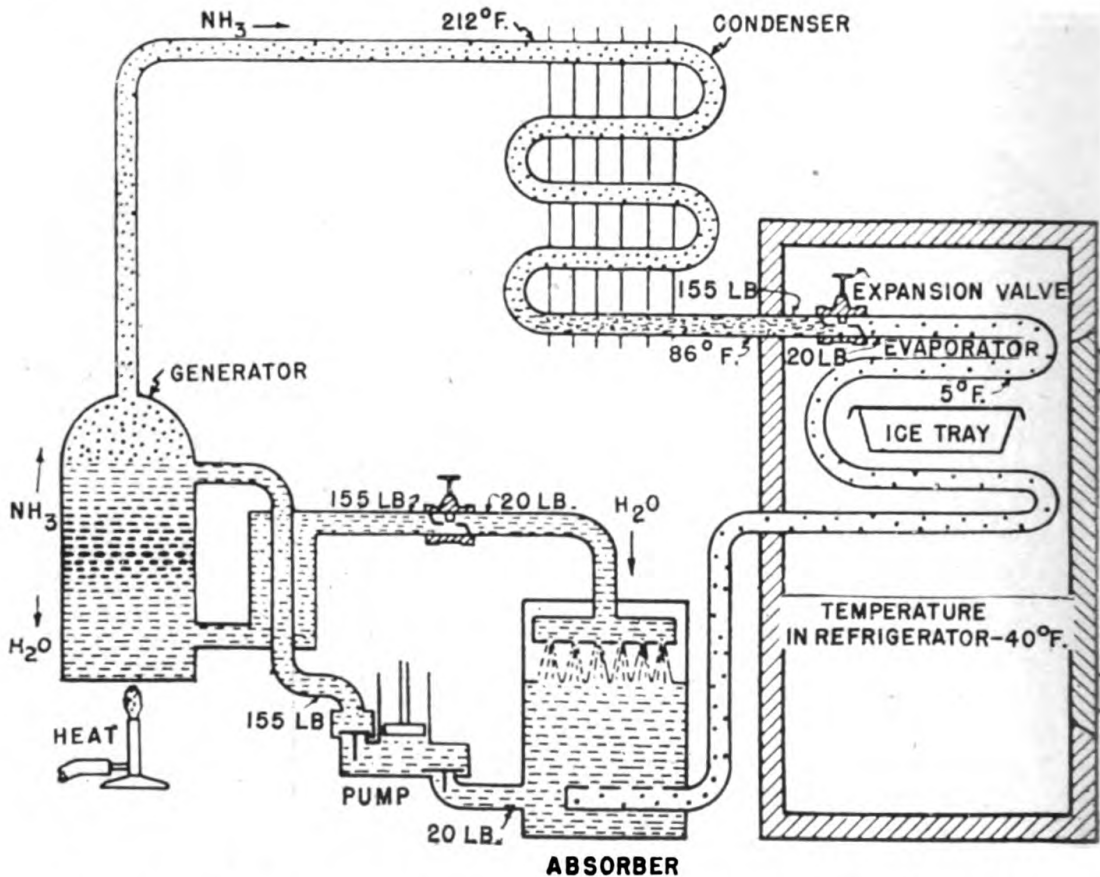


Fig. 1.3. A diagrammatic sketch of Ferdinand Carré's absorption system. From "Some Interesting Refrigeration Inventions," by Robert A. O'Leary. *Refrigerating Engineering*, Vol. 42, No. 5, November, 1941.

Figure 1.3 shows diagrammatically the apparatus which Ferdinand Carré introduced to the Confederate States from France during the Civil War. The need for refrigeration had become acute, since the supply of natural ice from the North had been cut off. This was the first heat-operated absorption system and consisted of an evaporator, condenser, generator, pump, and absorber. This unit was designed to operate with ammonia as the refrigerant and water as the absorbent.

A summary of the status of the science of refrigeration in 1876 is best given in an article by Dr. H. Meidinger of Karlsruhe, Germany.²

Concentrated cold in the form of ice acquired day by day a higher importance for industrial as well as domestic purposes. Brewing on the Bavarian system, the preparation of "Lager Beer," which amongst us in Germany, at least, has nearly superseded all other kinds of beer, depends upon the prolonged mainte-

² H. Meidinger, "Progress in the Artificial Production of Cold and Ice." *Journal Franklin Institute*, Vol. 101 (1876), p. 266.

nance of a temperature bordering upon the freezing point. The confectioner has no other practical means of producing a degree of cold from -12° to -18° , as required in the preparation of ice creams. The physician often employs the cold of ice both externally and internally as an absolutely indispensable remedy. The butcher and the hotel keeper can scarcely dispense with this means of preserving meat. In the domestic sphere ice has become formally established, at least in large cities, where it can always be obtained at a cheap rate, and to those who have become accustomed to its use, it appears a necessary agent for preserving food and cooling beverages during the warm season. In chemical manufacturers ice has also found various applications in the crystallization of salts, or, to speak in more general terms, in the separation of dissolved substances by means of cold. In proportion to the growing consumption, we see increasing quantities of ice stored up every winter. An extensive system of transportation has been arranged for conveying ice from the more northern and colder parts of this earth to regions near the equator. North America especially ships ice in astonishing quantities in all directions, even to Central and South America, to the West Indies and to India. Ice from Norway is sent to England and the German ports on the North Sea. In mild seasons, such as 1862-63 and 1872-73, ice from glaciers of the Alps was sent down the Rhine in entire trains.

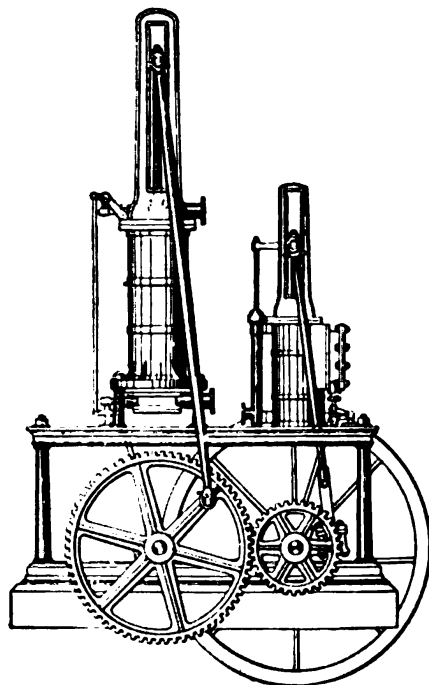


Fig. 1.4. James Harrison's sulfuric ether machine, with steam engine, 1859. Courtesy *Ice and Refrigeration*.

Science has shown, however, how to prepare the important requisite artificially. The first attempts at the manufacture of ice on the large scale took place between 1850-60; but this branch of industry has since been much extended. Even in regions where the winter is, as a rule, cold enough to permit ice to be stored up in quantity, e.g., in Germany, it has often been found remunerative to construct machinery for its artificial preparation, or in general terms, for the production of cold. Manufacturing establishments of this kind may be seen in various places in full activity, and after the mild winter of 1872-73, the demand for machine-made ice could scarcely be met.

Although it is evident that by this time applications of refrigeration were finding their way into the daily lives of the people, there were still some reputable engineers who were not enthusiastic about the future of mechanical refrigeration. In 1877 Dr. A. W. Hoffman³ wrote:

Ice machines, however they may be eventually improved and their effect increased, will never, in the more northern parts of the temperate zone, where a

³ A. W. Hoffman, "On the Development of the Chemical Art During the Last Ten Years." *Journal Franklin Institute*, Vol. 102 (1877), p. 135.

moderately cold winter with frost is generally experienced, acquire importance enough to meet the demand even approximately. They will serve merely as valuable substitutes to render us independent of the fickleness of the seasons.

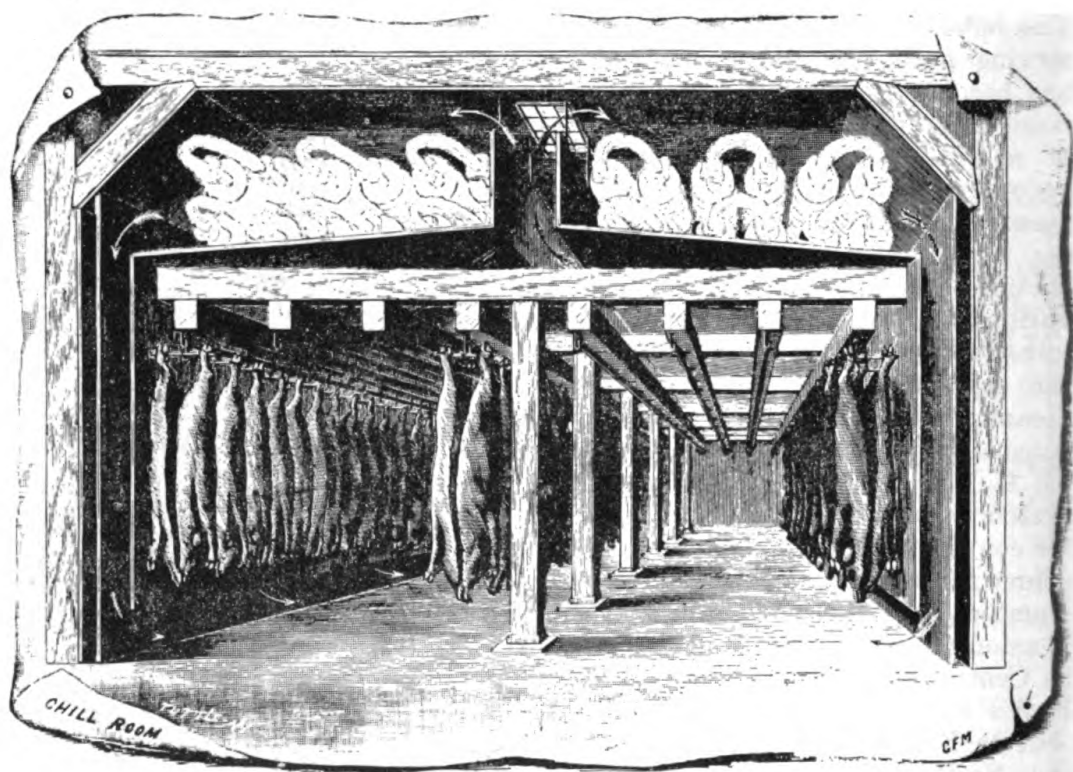
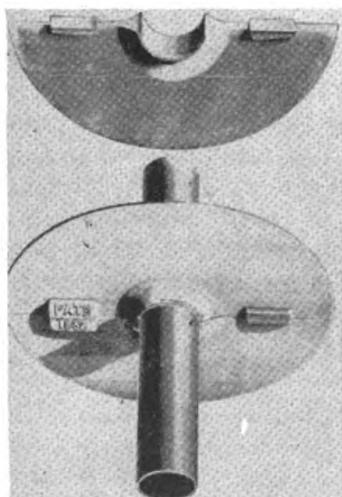


Fig. 1.5. Cast-iron disks served as fins on early cooling coils. (See view of fin shown below.) Woodcut from an 1890 De La Vergne catalog. Courtesy Frick Co.



At the beginning of the 1890's artificial refrigeration was coming into its own. Steam-jet and mechanical refrigeration systems were fairly common, and absorption refrigeration systems were receiving wide usage. However, many fallacious ideas were foisted upon the public, and many misconceptions prevailed. In 1892 it was aptly noted⁴ that many refrigerating schemes presented "... an order for the demolition of the second law of thermodynamics in the interest of a scheme for the dissipation of capital." In this respect times have changed little.

The early refrigeration industry was beset with difficulties. In 1888 six of the leading manufacturers held a secret meeting in New York to

⁴ George Richmond, "Notes on the Refrigeration Process and Its Proper Place in Thermodynamics," *Transactions, American Society of Mechanical Engineers*, Vol. 14 (1892-1893), p. 183.

discuss overproduction in their field. They seemed to feel that the solution was through consolidation, with two large firms dividing the country. The recently proposed Sherman Act, however, required that they proceed with caution. However, the whole scheme was abandoned during the following winter, when an ice-crop failure threw production back into high gear.

By 1900 a considerable portion of the industry's activity was consumed in the conversion from natural to artificial ice, although a battle over the relative merits of these two products raged for another 15 years. In the early 1900's any product that had been frozen, whether by nature or otherwise, was considered damaged beyond repair. A Brooklyn bakery was virtually ruined by publicity over its alleged use of frozen eggs in cake. Legislation threatened the very existence of cold-storage warehouses. Human lethargy was as difficult to overcome in those days as it is now.

Compressors were still driven by steam at the turn of the century, and the operating speeds of large units were kept well within 50 rpm. The huge size of these old units is probably as impressive as any single feature. Figure 1.7 shows a 220-ton⁵ De La Vergne refrigerating machine of about 1890. Figure 1.8 illustrates a 4-ton ammonia absorption system, and Fig. 1.9 illustrates one of the first domestic refrigerating units. Here the motive power was supplied by a hot-air motor or water turbine.

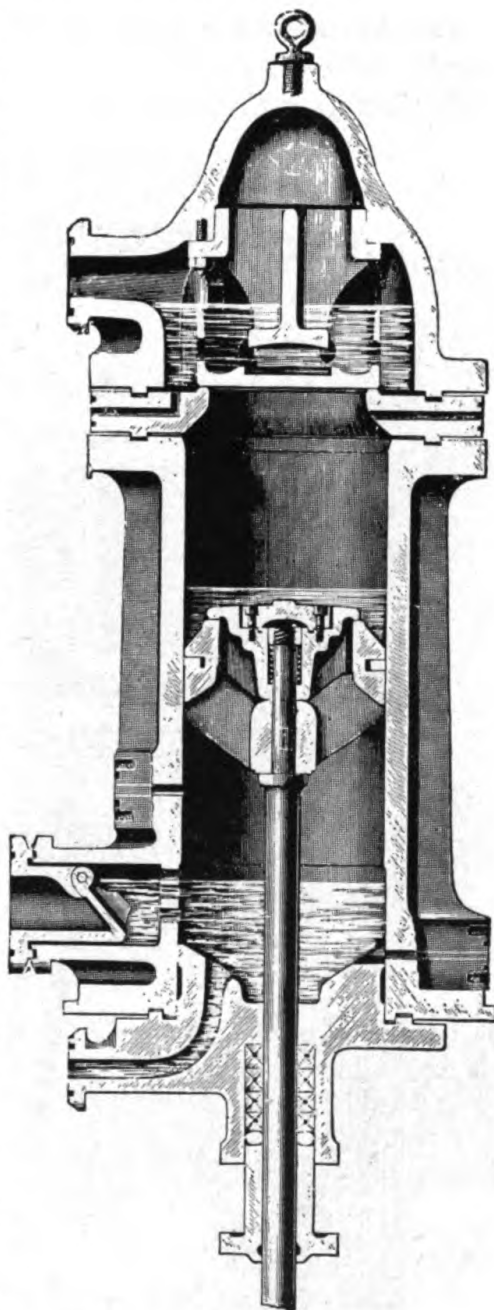


Fig. 1.6. Some early compressors injected the lubricant directly into the cylinder at end of return stroke. Thus, oil acted as a cooling medium in addition to its normal functions of lubrication and of sealing valves and piston rods. Woodcut from a De La Vergne catalog, 1890. Courtesy Frick Co.

⁵ A ton of refrigeration is defined as the removal of heat at the rate of 12,000 Btu per hour (see §2.2).

1.3. Refrigeration Early in the Twentieth Century. The Refrigerating Machinery Association was formed in 1903 and the American Society of Refrigerating Engineers in 1904. At this time ice-cream manufacture

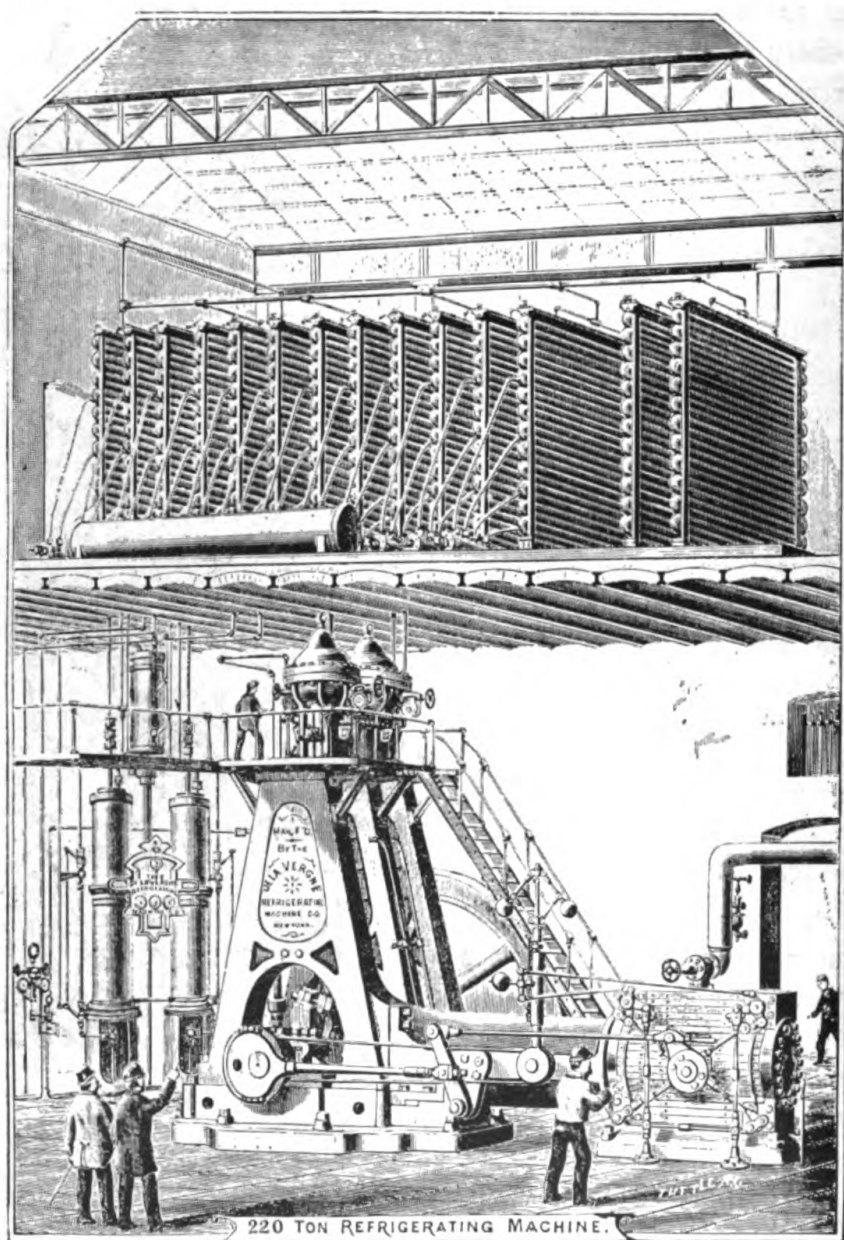


Fig. 1.7. A 220-ton De La Vergne refrigerating machine, 1890. Note the size of the condensers. Woodcut from an early De La Vergne catalog. Courtesy Frick Co.

was becoming an industry, and there were a few isolated applications of refrigeration for artificial skating rinks, fur storage, drinking water cooling, and air conditioning for the manufacture of camera film and bakery and candy goods. In 1905 Gardner T. Voorhees patented his famous multiple-effect compressor, in which the refrigerant gas from two different

evaporators at different pressures could be drawn in and compressed in a single cylinder. It is interesting to note that about 40 years later the development of a practical means for the adoption of this patent was just starting. A 450-ton cooling system for air conditioning the New York Stock Exchange had been installed by 1904, and a similar comfort cooling system was installed in a German theater at about the same time.

By 1911 compressor speeds had been raised to between 100 and 300 rpm, and in 1915 the first modern two-stage compressor was put in operation. Although many reputable engineers claimed it to be a dismal failure, it was still operating successfully in 1940. By the time of the First World War the National Bureau of Standards had made accurate determinations of the latent heat of ice. Rotary compressors and steam-jet units were receiving their initial impetus, and refrigeration was being commonly used in the oil-refining industry. At this time domestic refrigerating machines were practically unheard of, although a few of them had been purchased by the very wealthy. Even as late as 1919 a discussion between various members of the American Society of Refrigerating Engineers concluded that household units had few if any possibilities.

If there is a lesson to be gained in reviewing the history of an industry such as refrigeration, it is through analyzing in retrospect the heated scientific arguments of former years. No longer do we find that, as in the 1890's, soda water and ice cream form an excellent subject for Sunday sermons,⁶ that cold storage and artificial ice are unhealthful, or that

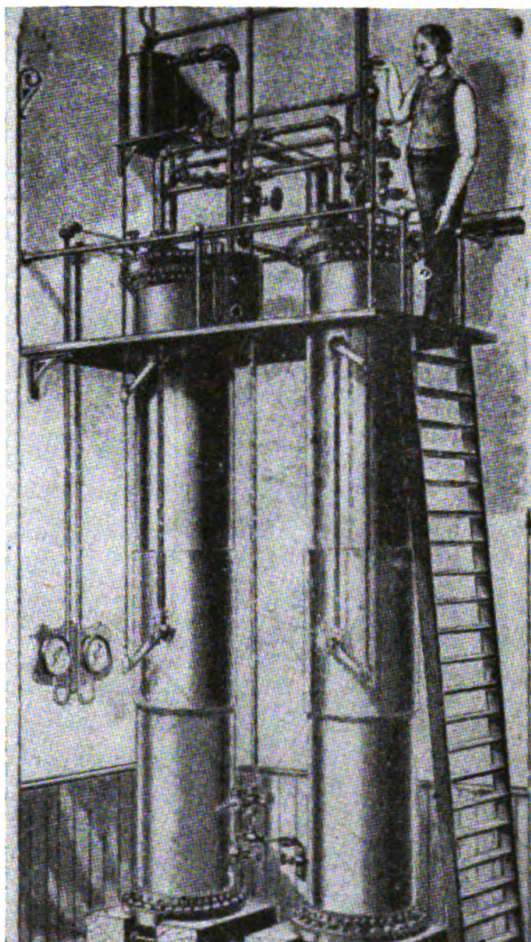
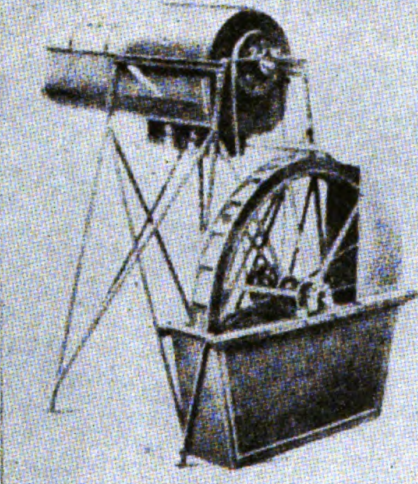


Fig. 1.8. An early four-ton ammonia-absorption machine. From "Refrigeration in the Gay Nineties," by David L. Fiske. *Refrigerating Engineering*, Vol. 40, No. 6, December, 1940.

⁶ According to Stewart Holbrook, "In time soda water and ice cream became excellent new menaces for the more excitable clergy. In the author's native town of Newport, Vermont, as late as 1890, a powerful sermon was preached against "sucking soda" and eating ice cream in drugstores on the Sabbath; and in certain Midwest

compressor speeds in excess of 100 rpm are well-nigh impossible. The rejection of a possible market for domestic refrigerating machines, and



Type "B"

This machine makes six to eight (6 to 8) pounds of ice in one hour. Motive power used on the machine illustrated is a water wheel. Price complete \$150.00

A Child Can Operate It

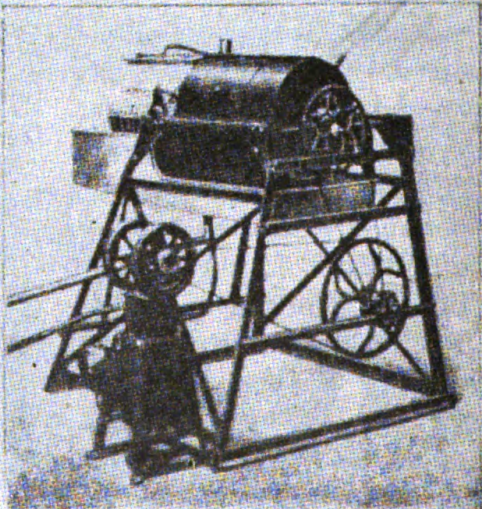
REMEMBER!

No acids, no chemicals
to buy or handle.

It is never necessary
to either take machine
apart or to re-change
the cylinder.

Requires a very small
amount of power
to run it.

The waste water is
sufficient or power
can be supplied by
means of an electric
or hot air motor or
water turbine.



Type "C"

This machine makes twelve to fifteen (12 to 15) pounds of ice in one hour, twenty-five minutes (1 hr. 25 min.). Motive power used on machine illustrated is a hot air motor. Price complete \$250.00

Fig. 1.9. Early domestic refrigeration units frequently used hot-air motors or water turbines. From "Refrigeration in the Gay Nineties," by David L. Fiske. *Refrigerating Engineering*, Vol. 40, No. 6, December, 1940.

of even a widespread use of refrigeration at all in the northern section of the country, now seems ludicrous. Two-stage and three-stage com-
towns laws were passed against the abomination, and the selling of soda water on Sunday was prohibited." *Lost Men of American History*. New York: The Macmillan Company, 1946, p. 120.

pressors are not only acceptable but necessary for low-temperature refrigeration, and air conditioning is an accomplished art.

1.4. The Industry Today. The prewar peak production year of the refrigeration and air conditioning industry was 1937. In the decade from 1927 to 1937 the expansion in dollar values of products manufactured was approximately 100 per cent. The total sales of the refrigeration and air-conditioning industry in 1937 were \$375,000,000. This may be compared with figures in the same year of \$718,000,000 for the heating-and plumbing-equipment industry and \$5,293,000,000 for the automobile industry.

In 1940 the total installed capacity of refrigerating equipment⁷ in this country was 6.8 million horsepower. If it is assumed that the various classifications of refrigerating equipment may be placed in four major divisions, then 39 per cent of this horsepower was for domestic refrigeration, 11.2 per cent for commercial refrigeration, 33.8 per cent in industrial refrigeration, and 16 per cent in air conditioning. In comparison with these figures the 1940 total capacity of all machinery of all kinds in industrial establishments in the United States was 51 million horsepower.

The expansion of the refrigeration industry in the last few years has been very great indeed. The world sales of household refrigerators in 1933 were 881,000 units. In 1941 this had leaped to 3.5 million, or an increase of roughly 400 per cent. Toward the end of 1946 they were being produced at about this same rate. In 1939 over 31 million tons of manufactured ice was produced, and in the same year some 357 million pounds of dry ice was produced by 44 establishments in this country. In 1940 approximately 14 million tons of food products was shipped via the railroads in refrigerator cars. In this same year there were 267 manufacturers of air-conditioning equipment, and the value of their sales at factory prices was over \$20,000,000. In 1941 there were 1879 cold-storage establishments in the United States, and 202 million pounds of quick-frozen fruits and 107 million pounds of quick-frozen vegetables were processed for consumption. Fifty thousand tons of frozen foods was shipped in refrigerated railroad cars in 1945 and it is predicted that 50 per cent of all perishable foods will be preserved by freezing before 1955. It is expected that the 5000 locker plants of 1943 will be increased to 20,000 by 1950.

In the last few years new industrial refrigeration applications have opened a comparatively new field for controlled-temperature application. In many plants air conditioning and refrigeration have become essential in the manufacture of products from both a quantity and a quality standpoint. Increased production of blast furnaces, as high as 27 per

⁷ All production and sales figures are from David L. Fiske, "The Demand for Refrigeration." *Refrigerating Engineering*, Vol. 46, No. 5 (November, 1943), p. 321.

cent, has been obtained by dehumidification of the air charged to the furnaces. Certain synthetic rubbers cannot be produced without access to temperatures in the range of -100°F . Air conditioning has been found useful in the manufacture of munitions, optical goods, plastics, textiles, smokeless powders, watches and other delicate instruments, and radio and electrical equipment, to name only a few applications. Low-temperature refrigeration is now used to obtain expansion fits as a substitute for shrink-fitting of metal parts. The low-temperature treatment of metals is a field virtually unexplored; one application is in the chilling of aluminum rivets to control age-hardening. The simulation of stratosphere conditions in the laboratory for the testing of all types of aircraft equipment, including the engines themselves, requires refrigeration at -67°F . In the process industries, refrigeration is used to carry out such operations as the separation of crystals from solutions. For example, phenol is frequently crystallized by refrigeration, and one borax-manufacturing plant alone requires several hundred tons of refrigeration for crystallization.

Applications of refrigeration in the medical profession are increasing daily, not only in the preservation of certain products but also in the actual treatment of some physical ailments. The use of "cold anesthesia" has shown great promise in the field of surgery. The manufacture of penicillin requires refrigeration and the removal of moisture under high vacuum. The same process is required in the drying of blood plasma, and the preservation of the blood requires refrigeration before the drying is accomplished. The refrigerant Freon-12 is widely used as an aerosol for the dispersal of DDT, the insecticide.

In the refrigerated foods industry developments are occurring so rapidly that it is difficult to keep abreast of them. Increased application of domestic refrigerators has been supplemented by the entirely new field of domestic low-temperature cabinets for the preparation and storage of frozen foods, both uncooked and precooked. Locker plants for the storage of frozen foods are becoming common in both rural and urban areas. Both low-temperature and high-temperature farm refrigeration form virtually a new outlet, following directly in the wake of rural electrification. Even the preservation of foods by dehydration requires, in many cases, that the products first be frozen and the moisture removed under high vacuum at low temperatures. The transportation of foods has received considerable attention in the last few years with new developments in the refrigeration of ships, railroad cars, and trucks.

These are but a few of the widespread applications of refrigeration. Present-day refrigeration requirements involve the entire temperature scale, almost down to absolute zero, with the lower temperature applications used in the laboratory for the study of basic thermodynamic and physical facts still incompletely understood. Many of these applications,

together with the design requirements for the equipment, are described in some detail in the later chapters of this text.

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CHAPTER 2

Basic Refrigeration Cycles and Concepts

2.1. Introduction. In the artificial production of low temperatures ranging from the normal ambient-air temperature down to absolute zero, one or more refrigeration cycles or processes may be involved. For cooling in the temperature range down to -200°F , *vapor-compression* refrigeration using *reciprocating*, *rotary*, or *centrifugal* compressors may be used. For some applications within this band, *air-cycle* refrigeration, *steam-jet* refrigeration, or *absorption* refrigeration may be applied. Such systems are commonly used for the production of cold, and their cycles are introduced in this chapter. For the production of extremely low temperatures down to absolute zero, irreversible or reversible *adiabatic* expansion of a gas, vaporization of a liquefied gas, or magnetic cooling may be used. A study of these specialized and limited processes is deferred until later in the text.

Although refrigeration is primarily an application of thermodynamics, many other phases of engineering are involved in the design, manufacture, application, and operation of refrigeration systems. Some knowledge of the chemistry of refrigerants is desirable in determining their reactions with metals and other materials with which they may come in contact. The thermodynamic properties of the refrigerants must be known before the cycle analyses can be made. A study of evaporators and condensers, those portions of the system used for the absorption and rejection of heat, respectively, involves the fields of heat transmission and fluid mechanics. In addition, steady state and, in some cases, periodic and transient heat transfer are involved in the determination of cooling-load requirements. The calculation of cooling loads also requires a knowledge of psychrometry. Fluid mechanics forms the basis for the sizing of the lines through which the gaseous and liquid refrigerants must flow. The design of reciprocating, rotary, or centrifugal compressors or of an air-cycle turbine involves a variety of machine design problems. The physical capacity of a compressor or expander will be determined from thermodynamic factors, but the physical design must also involve structural considerations. In the present chapter the basic refrigeration cycles and compressor modifications are introduced prior to their detailed analysis in subsequent chapters.

2.2. Standard Rating of Refrigerating Machines. The standard unit used in the rating of refrigerating machines, condensing units, and other

parts of a refrigerating system is the *ton of refrigeration*, defined as the removal of heat at the rate of 12,000 Btu per hour or 200 Btu per minute. The *ton-day of refrigeration*, also sometimes used, is defined as the heat removed by a ton of refrigeration operating for one day, or 288,000 Btu. These somewhat ambiguous terms have their origin in the concept of the amount of heat absorbed by a ton of ice when melting from the solid to the liquid phase at 32 F. This derivation assumes a latent heat of ice of 144 Btu per pound, whereas it is actually slightly less than this. However, no error is introduced in the calculations, because the units are defined in terms of heat removal and are no longer associated with the melting of a definite weight of ice.

2.3. Elementary Vapor-Compression Refrigeration Cycle with Reciprocating Compressor. At atmospheric pressure, liquid ammonia evaporates at -28°F (saturation temperature corresponding to 14.7 pounds per square inch absolute pressure or psia), and under these conditions one pound of liquid ammonia in changing to vapor absorbs 589.3 Btu (latent heat of evaporation). Thus the simplest form of a vapor refrigeration system consists of an open vessel containing a liquid refrigerant such as ammonia. The ammonia evaporates at temperatures below those surrounding the container and in so doing absorbs heat. However, such an uncontrolled refrigeration system is uneconomical, since the refrigerant is not recovered and the evaporating temperature is limited to that corresponding to the atmospheric pressure. Such systems are therefore never used commercially.

With proper auxiliary equipment, however, the refrigerant can be recovered and reused in a cyclic process; moreover, the temperature of evaporation of the refrigerant can be controlled by controlling the pressure. Thus if liquid ammonia is maintained at a pressure of 30.42 psia, the saturation or evaporating temperature is 0°F , and the latent heat of vaporization is 568.9 Btu per pound; if the absolute pressure is 59.74 psia, the evaporating temperature is 30°F , and the latent heat of vaporization is 544.8 Btu per pound. The refrigerant change from liquid to vapor or from vapor to liquid may be controlled by controlling the pressure of the refrigerant. If the ambient temperature surrounding the refrigerant and its container is above the saturation temperature corresponding to the refrigerant pressure, then evaporation, and consequently absorption of heat, takes place. If the refrigerant is already in the vapor state and if the temperature surrounding the refrigerant and its container is below the saturation temperature corresponding to the refrigerant pressure, condensation occurs. Other standard refrigerants serve as well as ammonia, and the pressure, volume, temperature, enthalpy, and entropy relationships may be determined from standard refrigerant tables.

A complete vapor-compression refrigeration system requires an *evaporator* to contain the boiling liquid refrigerant, a *compressor* to pump

the refrigerant vapor from the low-pressure to the high-pressure side of the system and to control the pressure within the evaporator, and a *condenser* for removing the heat from the refrigerant gas so that it may be returned to liquid form. In addition a *receiver* for storing the liquid refrigerant under high pressure and an *expansion valve* for controlling the rate of flow of the liquid refrigerant between the high- and low-pressure

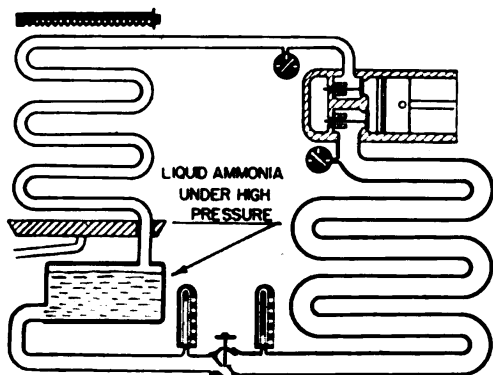


Fig. 2.1.

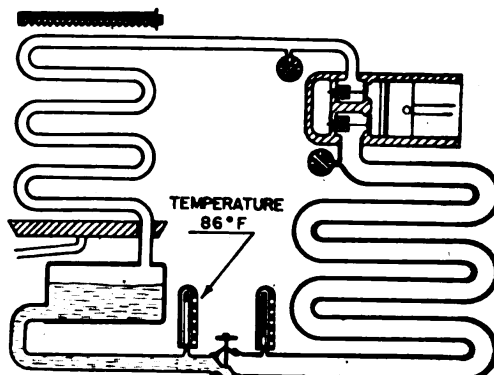


Fig. 2.2.

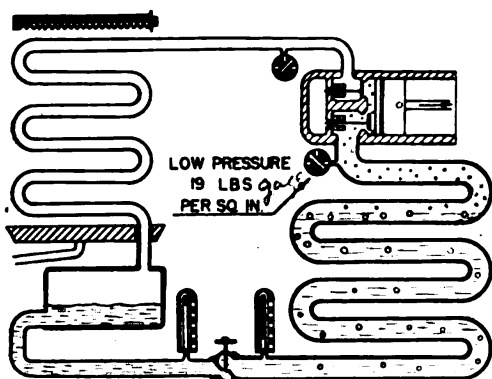


Fig. 2.3.

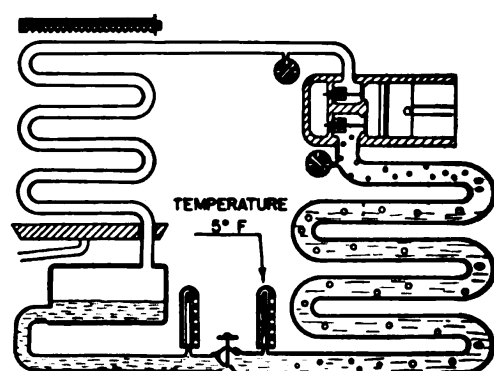


Fig. 2.4.

sides of the system are needed. Such a system is shown in progressive stages of operation in Figs. 2.1 through 2.8.

In Fig. 2.1 the high-pressure (154 psi gage) liquid ammonia is held in the receiver. Atmospheric pressure is assumed to be 14.7 psia.

In Fig. 2.2 the liquid passes to the entrance of the expansion valve. The temperature of the liquid ammonia is 86 F, the saturation temperature at 154 psi gage.

In Fig. 2.3 the liquid refrigerant is throttled through the expansion valve into the evaporator. Here a low pressure of 19 psi gage is maintained by operation of the compressor. The expansion valve shown is one in which the size of the needle opening is controlled by hand. In commercial operation, automatically controlled expansion valves are frequently used.

In Fig. 2.4 the liquid ammonia is evaporated at a temperature of 5 F corresponding to the evaporator pressure. The refrigerant is no

longer in a stable state, since the objects surrounding the evaporator are at a temperature higher than 5 F and thus supply the latent heat absorbed through the coil walls.

In Fig. 2.5 ammonia vapor from the low-pressure side of the system has been drawn into the compressor and discharged into the high-pressure side.

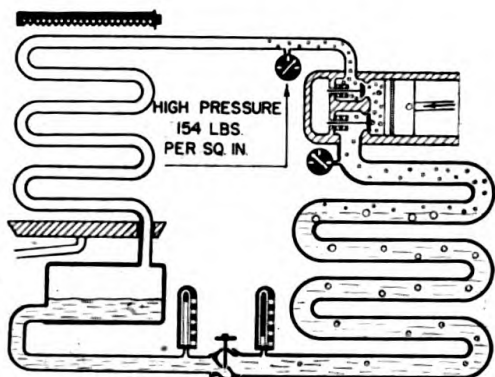


Fig. 2.5.

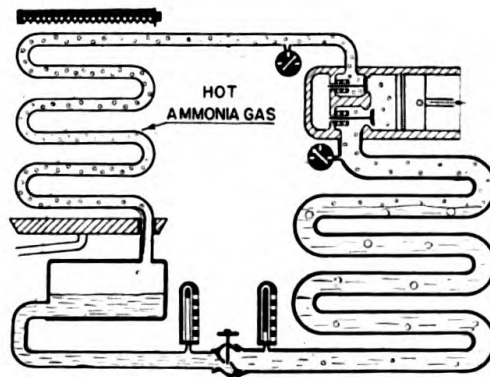


Fig. 2.6.

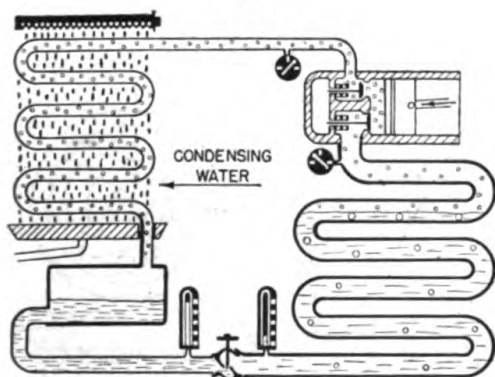


Fig. 2.7.

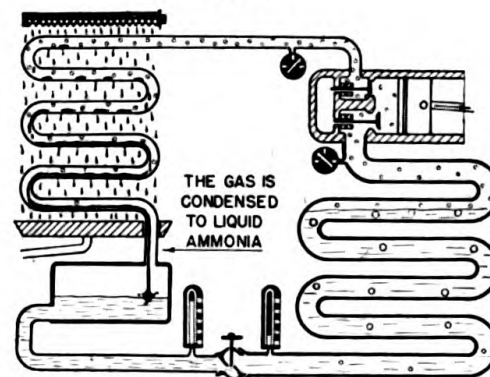


Fig. 2.8.

In Fig. 2.6 the high-pressure ammonia gas is discharged into the condenser. Both the condenser and the evaporator shown are simple but inefficient in design. More effective arrangements and designs for these heat-transfer surfaces are discussed in Chapter 11.

In Fig. 2.7 water passing over the condenser coils removes first the heat of superheat and then condenses the vapor by removing the latent heat. The heat removed by the condenser is equal to that absorbed in the evaporator plus the heat equivalent of the energy supplied to the vapor through the compressor.

In Fig. 2.8 the liquid ammonia from the condenser returns by gravity to the receiver. All processes occur simultaneously, only the action of the reciprocating compressor being intermittent in operation.

2.4. Elementary Vapor-Compression Refrigeration Cycle with Rotary Compressor. The vapor-compression refrigeration cycle may be applied to forms of equipment other than those discussed in §2.3. The com-

pressor need not be reciprocating, nor need the expansion device be an orifice with size controlled either by hand or automatically. The condenser may be air-cooled as well as water-cooled, and the evaporator may be flooded with the liquid refrigerant. Such a modification with a rotary compressor, flooded evaporator, capillary expansion tube, and air-cooled condenser, all similar to those used in domestic refrigerators, is shown in progressive stages of operation in Figs. 2.9 through 2.14.

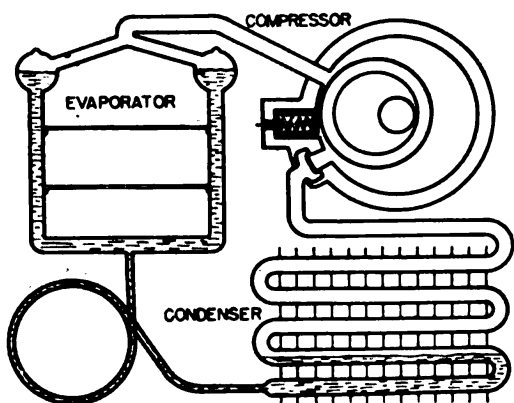


Fig. 2.9.

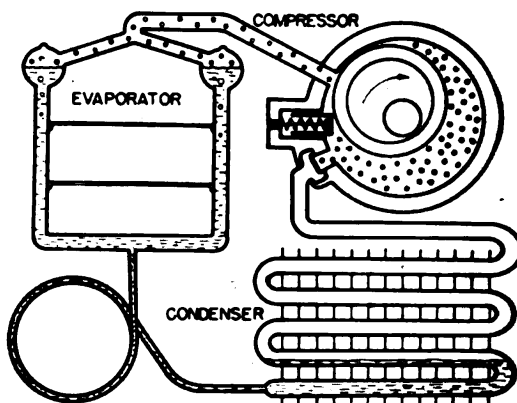


Fig. 2.10.

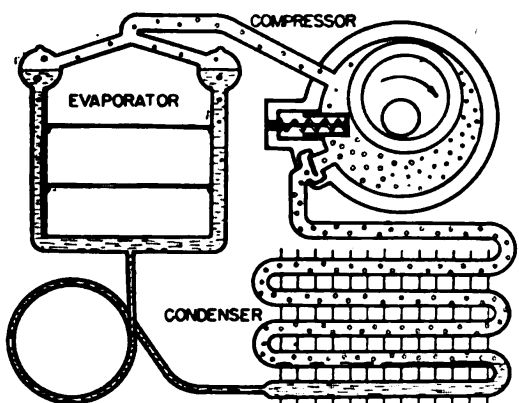


Fig. 2.11.

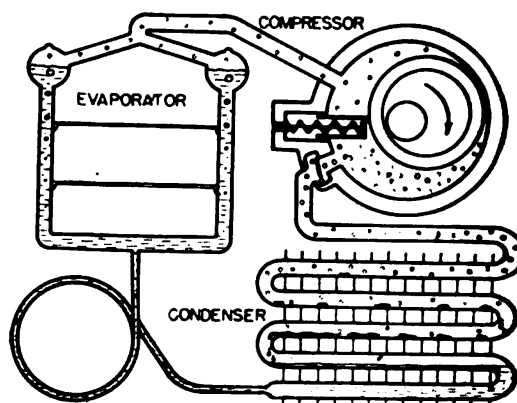


Fig. 2.12.

In Fig. 2.9 the system is inoperative, with the liquid refrigerant stored in the evaporator, the capillary tube, and the condenser. Since the evaporator is at a lower temperature than the condenser, the greater part of the liquid refrigerant is stored in that portion of the system. The remaining space in the system contains gaseous refrigerant. Pressures in all parts of the system are equalized.

In Fig. 2.10 the rotary compressor has started revolving, and some heat has entered the evaporator, releasing some low-pressure refrigerant vapor. The design of rotary compressor shown consists of an eccentric-driven rotor revolving within a closed housing and with a spring-operated sealing blade separating the suction and discharge passages. In the position of the rotor shown, the suction space between the eccentric and

the blade is very small, and the compression chamber volume is close to a maximum.

In Fig. 2.11 the suction chamber has increased in volume, and the discharge chamber has decreased in volume. Compression of the gaseous refrigerant in the discharge chamber has progressed to a point where it is slightly higher than the condensing pressure, and the discharge valve has opened, permitting a flow of gas from the compressor to the condenser.

In Fig. 2.12 continued movement of the rotor has increased the suction space, thereby increasing the volume of the vapor charge from the evaporator. The discharge space has been further reduced, and flow of gas to the condenser continues.

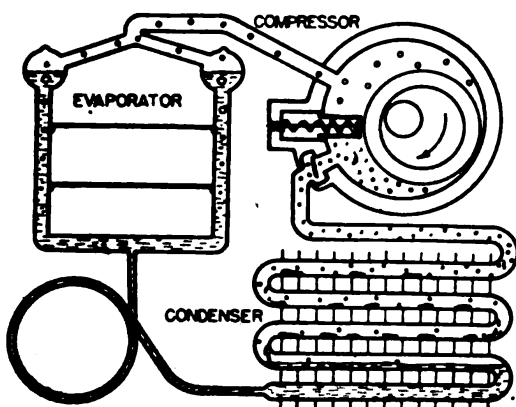


Fig. 2.13.

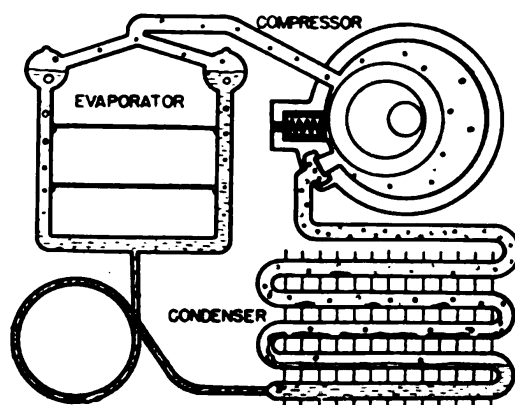


Fig. 2.14.

In Fig. 2.13 the rotor has progressed further. It is assumed that a sufficiently large number of rotor revolutions have occurred so that the charge of refrigerant entering the condenser has started to condense. The condenser is air-cooled and has finned exterior surfaces. Movement of air over the condenser may be by either gravity or forced convection.

In Fig. 2.14 the compressor rotor is shown in the dead-center position, with all the compressed refrigerant discharged. The condenser pressure has closed the discharge valve, and it will remain in that position until another revolution compresses the next suction charge to the condenser pressure. The flow of the liquid refrigerant from the condenser to the evaporator has started through the capillary tube. The restriction to flow caused by this small-bore tube is sufficient to allow the pressure in the condenser to build up to the point where condensation can occur with air as the cooling medium. As liquid refrigerant passes through the capillary tube, a continuous drop in pressure occurs, with a partial flashing of the liquid to vapor. The amount of liquid flashed becomes progressively greater as the refrigerant travels through the tube and is a maximum at discharge into the evaporator. Here as much as

20 to 30 per cent of the liquid refrigerant may already be in gaseous form, dependent upon the drop in pressure from the condenser to the evaporator and upon the amount of liquid refrigerant subcooling that has occurred in the condenser.

2.5. Elementary Vapor-Compression Refrigeration Cycle with Centrifugal Compressor. The vapor-compression refrigeration cycle may also be adapted to equipment designed for *centrifugal compression* of the refrigerant vapor. With such a system compression is accomplished

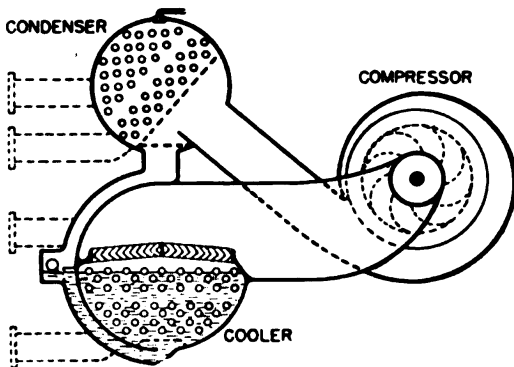


Fig. 2.15.

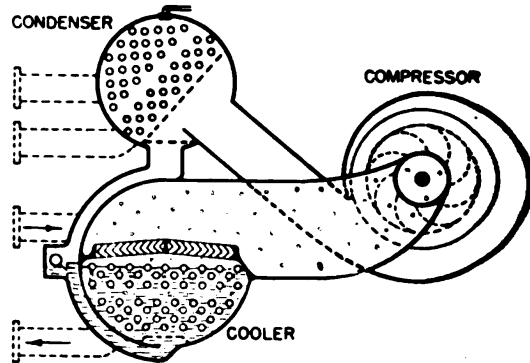


Fig. 2.16.

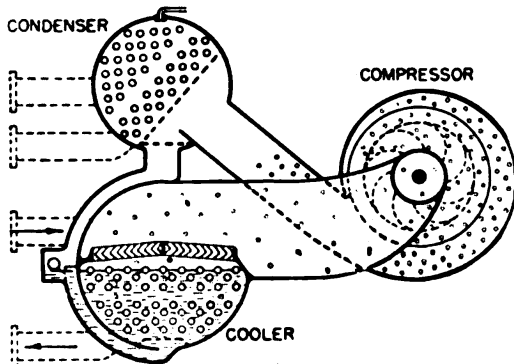


Fig. 2.17.

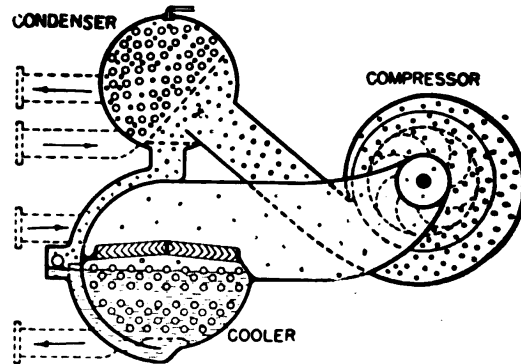


Fig. 2.18.

through centrifugal force in a manner similar to that utilized in the movement of air by a centrifugal blower. Since compression is not positive, the pressure differential developed is low compared with that for reciprocating compressors. Therefore, refrigerants operating satisfactorily with low compression ratios must be used.

Centrifugal compression systems are usually designed with a *condenser*, *evaporator*, and *compressor*, all mounted on a single base. A *secondary refrigerant* such as a *brine* or *water* is used to convey heat from the point of the load to the evaporator. Progressive stages in the operation of a typical centrifugal refrigeration system are shown in Figs. 2.15 through 2.19.

In Fig. 2.15 the system is inoperative, with the liquid refrigerant stored in the cooler or evaporator. The liquid refrigerant surrounds the cooler tubes, through which brine is circulated when the system is under load.

In Fig. 2.16 the cooler is loaded through the circulation of brine from the point of the load to the cooler tubes. Heat is absorbed, and boiling of the liquid refrigerant occurs. The centrifugal compressor is started, thus removing refrigerant vapor from the cooler and at the same time evacuating the cooler to an evaporator pressure corresponding to the desired evaporator temperature. Refrigerant vapor is shown entering at the hub of the centrifugal compressor.

In Fig. 2.17 refrigerant vapor has passed through the compressor and is shown entering the passageway leading to the condenser. Compression of the refrigerant has occurred by virtue of the centrifugal force imparted to the vapor by passage through the compressor impeller and by the additional velocity head imparted to the mass as it is carried in rotation by the whirling impeller.

In Fig. 2.18 the gaseous refrigerant has entered the condenser and surrounds the condenser tubes, through which cold water has started to circulate.

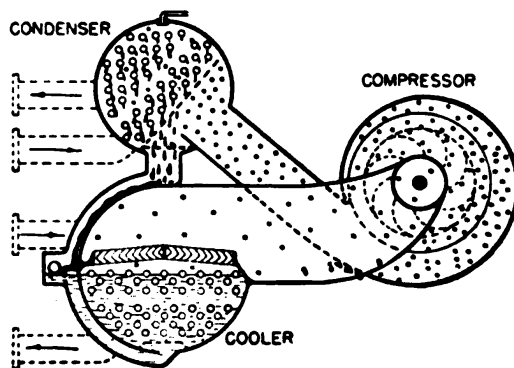


Fig. 2.19.

In Fig. 2.19 condensation of the refrigerant is occurring in the condenser, and the liquid flows by gravity to the cooler. A constant level of the refrigerant in the cooler is maintained by means of a float-operated valve. The system is now in complete and continuous operation.

2.6. Elementary Ejector Refrigeration System. Another variation of the basic vapor-compression refrigeration cycle is obtained by using an *ejector* to remove the refrigerant vapors from the evaporator chamber and to control the evaporator pressure. In such a system the fluid used to operate the ejector is mixed with the evacuated refrigerant vapors, and hence the design is greatly simplified if the motive fluid used for operation of the ejector is the same as the refrigerant. Such a system, therefore, is inherently adaptable to the use of water as the refrigerant and steam as the ejector fluid, and when so designed is termed a *steam-jet refrigeration system*. The equipment required is an *evaporator chamber*, a *primary steam ejector* and one or two *secondary ejectors*, a *primary condenser* and one or two *secondary condensers*, and two *liquid pumps*. Such a system is shown in progressive stages of operation in Figs. 2.20 through 2.24.

In Fig. 2.20 the steam-jet refrigeration system is inoperative, with water stored in the evaporator but not circulating through it.

In Fig. 2.21 steam at high velocity passes through the primary nozzle and through the ejector venturi leading to the primary condenser. Any vapors in the evaporator are entrained with the primary steam jet, and a vacuum is pulled upon the evaporator chamber. For efficient operation the motive steam must be supplied at approximately 100 to 150 psia.

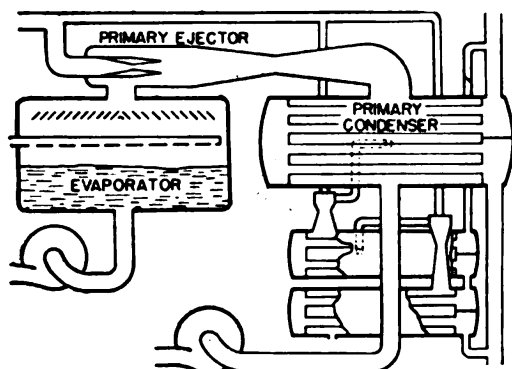


Fig. 2.20.

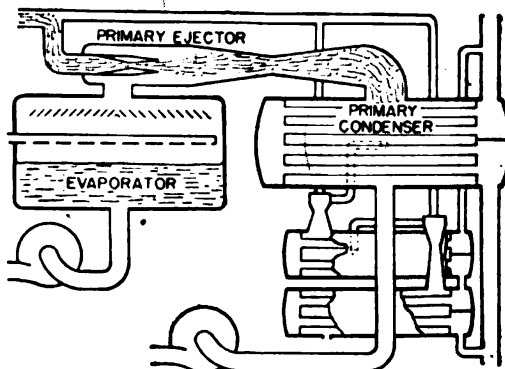


Fig. 2.21.

In Fig. 2.22 water, acting as both a primary and a secondary refrigerant, is circulated through the evaporator or flash chamber. The water returned from the point of the refrigeration load is sprayed through nozzles into the flash chamber in order to provide exposure of the maximum surface area for rapid evaporation. The evaporated water passes through

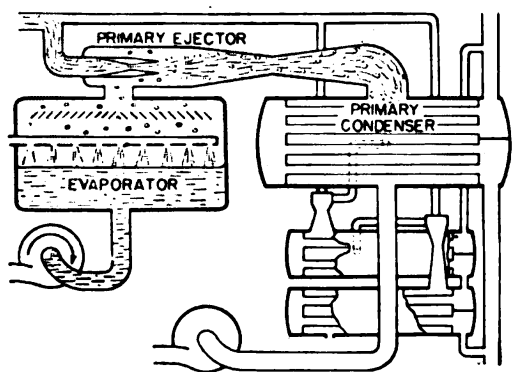


Fig. 2.22.

eliminator plates for removal of any entrained liquid and hence into the primary ejector stream. The ejector is capable of a compression ratio of approximately 8 to 1. Therefore, if the evaporator pressure is 0.25 in. of mercury absolute (corresponding to a 40 F evaporation temperature), the pressure in the primary condenser will be approximately 2 in. of mercury absolute. Refrigeration is accomplished by the flashing of a portion of the circulated water to water vapor, with the latent heat of vaporization absorbed from the remaining mass of water. Thus the water leaving the evaporator is several degrees lower in temperature than that entering and may therefore be used for refrigeration. Make-up water is added to the system to take the place of the vapors flashed and removed from the evaporator.

In Fig. 2.23 the motive steam, together with the flashed water vapor, has entered the primary condenser. Condenser cooling water is also

flowing, and both the motive steam and the flashed vapor are recovered by condensation. Since this water is at a low pressure (2 in. mercury absolute), it is returned by forced circulation to the boiler supplying the motive steam.

In Fig. 2.24 the secondary condensers and ejectors are placed in operation. These enable noncondensed vapors and gases to be removed from the primary condenser and, by two successive stages, to be raised to atmospheric pressure. The noncondensable gases are discharged to the atmosphere from the final condenser. The system is now in complete and continuous operation.

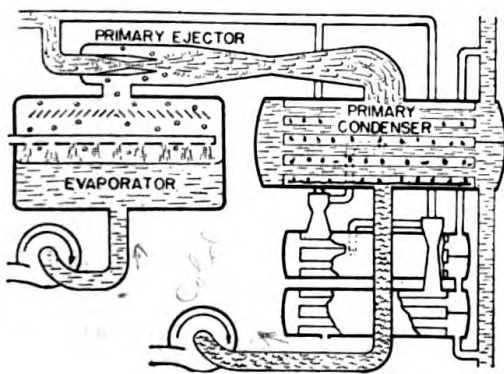


Fig. 2.23.

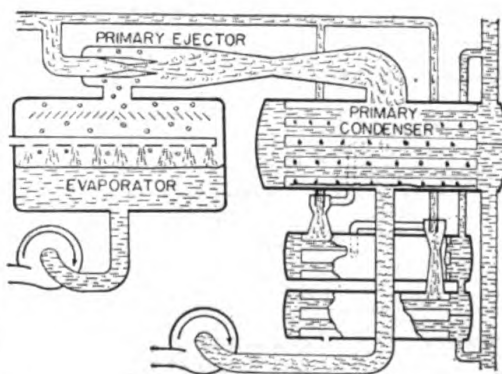


Fig. 2.24.

2.7. Elementary Air-Cycle Refrigeration System. The *air-cycle* refrigeration system is unique in that it is the only cooling process ever used widely in commercial application in which the refrigerant remains gaseous throughout the cycle. Because air is free of cost and completely safe, it is used as the refrigerant. An air-cycle system may be either *open* or *closed*. In a closed system the air refrigerant is confined to the piping and component parts of the system at all times; in the open system the air is expanded directly into the space to be cooled, allowed to circulate throughout the cooler, and is then returned to the compressor to start another cycle. Air-cycle refrigeration systems as originally designed and installed are now practically obsolete because of their inherently high horsepower requirements. However, modern adaptations to aircraft refrigeration have proved to be desirable because of the low weight and volume of the equipment and the ability of the system to utilize the available cabin-air supercharging system.

An air-cycle refrigeration system requires a *compressor*, a *cooler*, an *expander*, and a *refrigerator*. Because the refrigerant remains in the gaseous state at all times, the terms "condenser" and "evaporator" have no true meaning and are replaced by "cooler" and "refrigerator," respectively. Figures 2.25 through 2.29 show progressive stages in the operation of an elementary air-cycle refrigeration system.

In Fig. 2.25 an inoperative closed air-cycle system is shown. The compressor is a conventional reciprocating single-acting compressor with spring-actuated poppet valves, and the expander is of similar design. Both compressor and expander are attached to a common reciprocating shaft connected to an outside source of power, which may be a steam, electric, or internal-combustion engine.

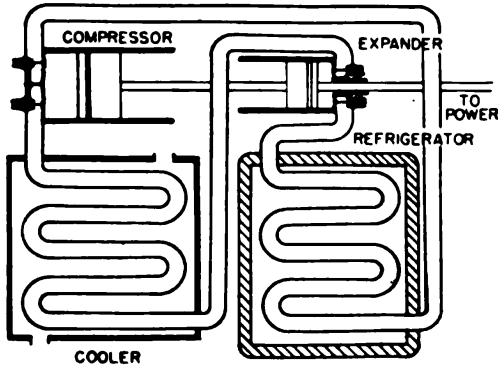


Fig. 2.25.

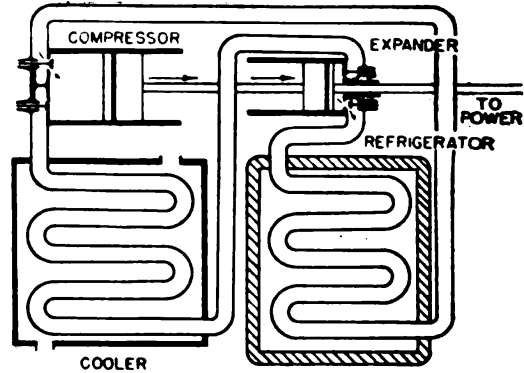


Fig. 2.26.

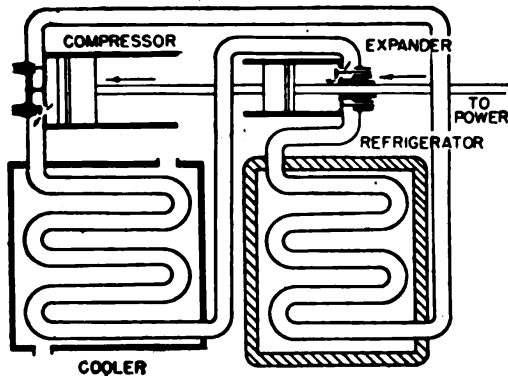


Fig. 2.27.

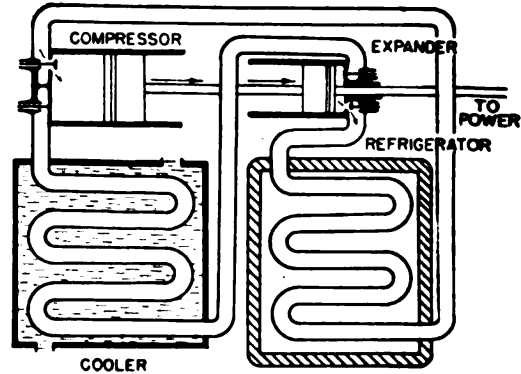


Fig. 2.28.

In Fig. 2.26 external power is applied to the reciprocating shaft, and a charge of air is drawn into the compressor. At the same time the charge of air already in the expander cylinder is exhausted through a discharge valve to the refrigerator.

In Fig. 2.27 the compressor charge is compressed and discharged to the cooler. At the same time a charge of compressed air from the cooler is allowed to pass into the expander cylinder, and the work of expansion is recovered and aids in operation of the compressor. After the system has been in operation for a short time, the cooler will be charged with compressed air at pressures of approximately 150 to 200 psia. The refrigerator will be charged with air expanded from high to low pressure and at a pressure of approximately 15 to 30 psia. Thus the section of the system from the pressure discharge valve through the cooler and to the entrance of the expander will be under high pressure, and the section

from the discharge of the expander through the refrigerator and to the entrance of the compressor will be under low pressure.

In Fig. 2.28 the compressor and expander valves are again shown in the same positions as Fig. 2.26. Water is now flowing through the cooler to remove the sensible heat resulting from compression of the air. The compressed air leaving the cooler and entering the expander will be at temperatures of about 70 F.

In Fig. 2.29 the compressor and expander valve positions duplicate those of Fig. 2.27. The approximately isentropic expansion of the compressed air into the refrigerator will result in an appreciable lowering of the air temperature, with the discharged air in some cases as low as -100 F. Frost formations have now appeared on the refrigerator coils, and the system is in complete and continuous operation.

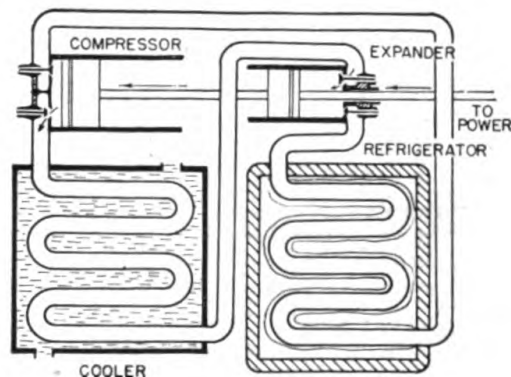


Fig. 2.29.

2.8. Elementary Absorption Refrigeration Cycle. In an *absorption* refrigeration cycle the compressor is replaced by an *absorber* and a

generator. The system operates essentially upon the standard vapor-compression refrigeration cycle, but a *secondary fluid* is introduced in addition to the refrigerant. In an *absorbing chamber* the refrigerant vapor from the evaporator comes in contact with and is absorbed by this secondary fluid or *absorbent*. This strong absorbent (high in concentration of absorbed refrigerant) is then transferred to the generator, where heat is applied, and the refrigerant vapor is driven off and into the condenser at high pressure. The resulting weak absorbent (weak in concentration of refrigerant) is then returned to the absorber, where the cycle is repeated. Figures 2.30 through 2.34 present progressive stages in the development of an absorption refrigeration system.

In Fig. 2.30 the simplest form of refrigeration by evaporation is represented by an evaporator coil passing through an insulated space to be refrigerated. Alcohol, ether, or some other readily evaporated liquid is allowed to flow in at the top of the evaporator, and a fan is supplied to create a steady flow of air over the evaporating liquid. The current of air removes the gaseous refrigerant, and the latent heat of vaporization is supplied through the evaporator-coil walls. Such a system is unsatisfactory, since the evaporating temperature is uncontrolled and must correspond to the saturation temperature of the refrigerant at atmospheric pressure. It is also uneconomical, because the refrigerant is not recovered.

In Fig. 2.31 several improvements have been added to the elementary

system of Fig. 2.30. The evaporator has been extended into a complete circuit, and an absorber has been added. The refrigerant is alcohol. The absorber water is poured into the system and is allowed to flow over a bend in the tubing so that the evaporated alcohol must flow through a curtain of water. Here absorption of the alcohol vapor occurs, and the absorbent becomes a water-alcohol mixture. The fan is eliminated from the system, since the cold vapors leaving the refrigerator are greater in density than the comparatively warm air leaving the evaporator. In the process of absorption some heat is released, thereby warming this air. This system is still impractical, however; for although the refrigerant is

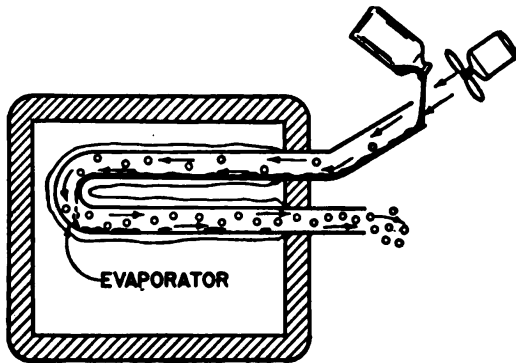


Fig. 2.30.

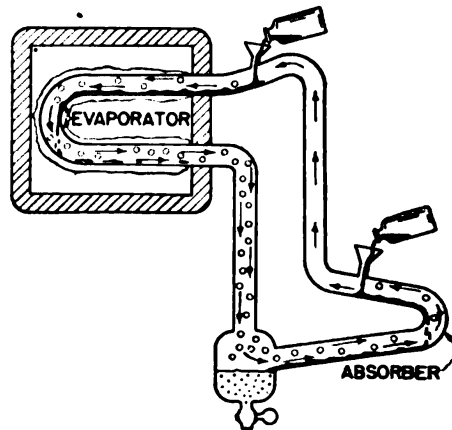


Fig. 2.31.

recovered, it has not been returned to a usable state. Furthermore, the alcohol-water solution collecting in the chamber at the bottom of the absorber must be constantly removed from the system.

In Fig. 2.32 further improvements have resulted in a practical absorption refrigeration system. Here the strong solution of alcohol and water is drained by gravity to a generator. Heating of the strong absorbent drives off much of the alcohol from the solution, and a *bubble pump*, operating upon the ordinary coffee-perculator principle, drives alternate slugs of liquid and alcohol vapor into a *separator*. From the separator the alcohol vapor flows to the condenser, where liquid alcohol is formed for reuse as a refrigerant. The weak absorbent solution drains by gravity from the separator into the absorber, where the liquid again picks up the alcohol vapor from the evaporator. Liquid seals are necessary in both the discharge from the condenser and the discharge from the separator.

Commercial adaptations of the cycle shown here are usually limited to fractional and small-tonnage systems, because a bubble pump is inadequate for movement of large volumes of liquid. Ammonia is commonly used as a refrigerant and water as the absorbent. Hydrogen gas is substituted for air as a conveying medium in the evaporator and absorber coils. In order that the system may operate properly with a refrigerant

such as ammonia, it is necessary that the pressure of the evaporating ammonia be much lower than that of the condensing ammonia. This low temperature is obtained by proportioning the volumes of water, ammonia, and hydrogen so that the required pressure of the evaporating ammonia plus the pressure of the hydrogen gas in the evaporator-

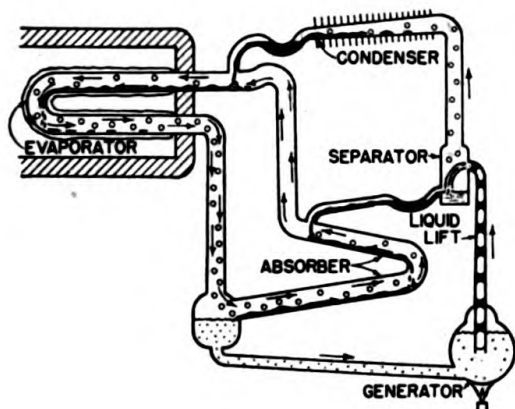


Fig. 2.32.

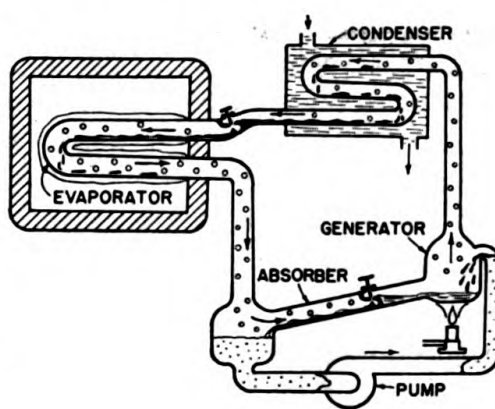


Fig. 2.33.

absorber circuit is equal to the comparatively high pressure of the condensing ammonia alone in the generator-condenser portion of the system. Thus, by segregating the hydrogen gas in the evaporator-absorber circuit, the pressure differences between the parts of the system are small, and mechanical moving parts are eliminated.

In Fig. 2.33 a simplified commercial adaptation of the cycle as applied to large systems is shown. Hydrogen gas has been eliminated from the evaporator-absorber circuit, and now it contains only low-pressure evaporating ammonia. This condition makes two expansion valves necessary: one between the condenser and the evaporator and the other between the generator and the absorber. A mechanical liquid pump is required to transfer the strong absorbent from the low-pressure absorber chamber into the high-pressure generator chamber. A water-cooled condenser has replaced the air-cooled condenser of the previous system. Absorption units developed from this elementary design are manufactured with capacities as high as several thousand tons.

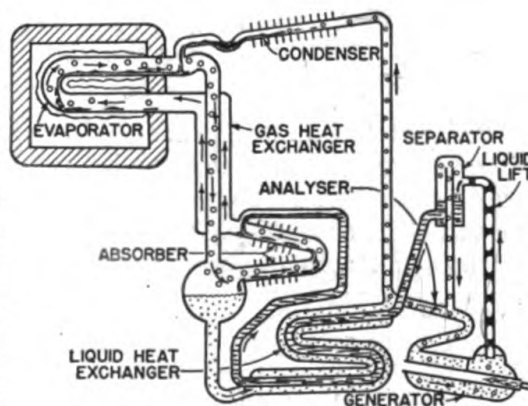


Fig. 2.34.

Figure 2.34 shows a refined commercial adaptation of Fig. 2.32 as applied to domestic refrigerators. The only additions over the basic

cycle are a *gas heat exchanger*, a *liquid heat exchanger*, and an *analyzer*, all added for greater efficiency. The liquid heat exchanger is used to heat the strong absorbent entering the generator while cooling the weak absorbent entering the absorber. The gas heat exchanger cools the hydrogen entering the evaporator while heating the evaporated ammonia entering the absorber. A small amount of the vapor leaving the separator is condensed in passage through the strong absorbent in the analyzer. Since some water vapor is carried along with the ammonia gas leaving the separator and since the first gases to condense are strong in water vapor if any is present, the remaining ammonia gas entering the condenser becomes concentrated. This procedure also prevents sufficient water from entering the evaporator and forming frost.

PROBLEMS

2.1 A creamery must cool 2500 gal of milk received each day from an initial temperature of 80 F to a final temperature of 38 F in 3 hr. If all losses are neglected, what must be the capacity of the refrigerating machine? The density of milk is 8.6 lb per gallon, and the specific heat is 0.935.

2.2 If the compressor in Problem 2.1 is to operate 8 hr per day with the surplus refrigeration accumulated in brine-storage tanks, what must the capacity be?

2.3 Five hundred pounds of beef is to be cooled from 75 to 0 F in 6 hr. Freezing occurs at 28 F, the specific heat before freezing is 0.77, the specific heat after freezing is 0.40, and the latent heat of fusion is 100 Btu per pound. Determine the required capacity of the refrigerating machine.

2.4 A cold-storage room has inside dimensions of 40×25 ft with a 10-ft ceiling. The over-all coefficient of heat transmission [Btu per (sq ft)(F)(hr) from air to air through the construction] for the walls and ceiling is 0.09 and for the floor is 0.30. The inside temperature is to be maintained at 25 F. If the outside-air temperature adjacent to walls and ceiling is 80 F and that adjacent to the floor is 60 F, determine the capacity of refrigerating unit required for 70 per cent running time.

2.5 Fifteen hundred gallons of ice cream is to be manufactured per day. The ice-cream mix, which has a specific gravity of 1.10, swells 55 per cent during the freezing process. Determine the tons of refrigeration required to accomplish the initial freezing from 45 F to 22 F if the heat absorbed per pound is 60 Btu (latent and sensible). Freezing must be accomplished in 6 hr, and refrigeration losses are 10 per cent.

2.6 Cold salt brine at an initial temperature of 30 F is used in a packing plant to chill beef from 100 to 38 F in 18 hr. Determine the gallons of brine required to cool 1000 beeves of 500 lb each if the final brine temperature is 35 F. The specific gravity of the brine is 1.05 and the specific heat 0.90. The specific heat of the beef is 0.75.

CHAPTER 3

Review of Thermodynamics

3.1. Introduction. Refrigeration is concerned with the absorption of heat from a location where it is objectionable plus its transfer to and rejection at a place where it is unobjectionable. Regardless of the means by which this heat transfer is accomplished, the problem is one of applied thermodynamics. In some methods of refrigeration the working medium approaches a perfect gas, and the relatively simple thermodynamic equations for perfect gases may be applied. In the more commonly used forms of refrigeration, however, the working medium or refrigerant alternates between the liquid and vapor phases. Here the thermodynamic relationships are complex, and it is necessary to use tables and charts to present the physical and thermodynamic properties. This chapter is concerned with summarizing the important relationships of both gas and vapor thermodynamics, with emphasis on those phases most applicable to refrigeration.

3.2. Ideal Gases and Vapors. An *ideal* or *perfect* gas is one that obeys the relationship

$$pv = RT \quad (3.1)$$

and the other perfect gas laws exactly and has constant specific heat. Here p denotes pressure, v specific volume, R the gas constant, and T absolute temperature. The volume of such a gas will be zero at absolute zero, and its molecules will possess pure rectilinear motion between contacts. In order that this may be the case, such a gas must be entirely free from any molecular forces. The specific heats of such a gas are constant regardless of pressure or temperature changes, and, since the molecules may possess no energy of position, it is impossible that the gas have any internal latent heat or heat of disgregation. Hence it is impossible to change such a gas to the liquid or the solid states by lowering the temperature and increasing the pressure.

No real gases follow this behavior prescribed for ideal gases. In cooling a real gas toward absolute zero, it is eventually condensed and finally solidified, thus indicating departure from the perfect thermodynamic relationships. Although no real gas conforms exactly to equation 3.1, all experience indicates it to be the limiting relationship as the pressure approaches zero or the temperature infinity. If the conditions under which a gas exists, however, are sufficiently removed from its

critical temperature and pressure, equation 3.1 and the other perfect gas laws apply with sufficient accuracy for all practical purposes.

A substance existing in the gaseous phase but relatively near its saturation temperature is usually termed a *vapor*. Under such conditions the gas laws no longer apply, the degree of error increasing with proximity to the liquid phase. Figure 3.1 shows isotherms plotted on a pressure-volume diagram for a substance in the liquid, vapor, and gaseous states. In the gaseous region equation 3.1 is applicable and, when the temperature is held constant, reduces to the form $pv = \text{a constant}$, the equation

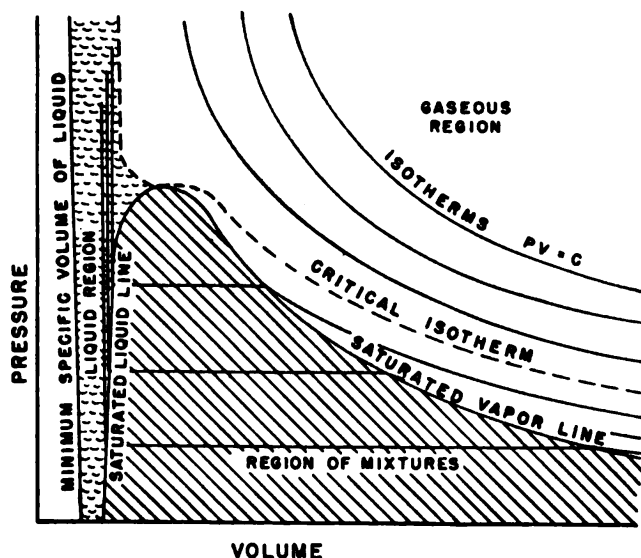


Fig. 3.1.

of a rectangular hyperbola. As the isotherms approach the *critical isotherm* defining the temperature above which the substance cannot exist as a liquid, the deviation from this equation becomes apparent. It is in this region that gases are usually termed vapor, and most refrigerants are in this region when in the vapor state. In this and in the liquid regions many equations of state have been proposed in substitution for equation 3.1, but none of reasonable complexity have sufficient precision for practical purposes. The most historical of these is van der Waals' equation:

$$\left(p + \frac{a}{v^2}\right)(v - b) = RT \quad (3.2)$$

in which a and b are constants varying with the gas. The constant a is introduced to compensate for the intermolecular forces in the real gas and the constant b to compensate for the reduction in specific volume due to the volume of the molecules.

3.3. Properties and State of a Substance. A property of a substance is any quantitative characteristic, a difference of whose values between

two states of the substance is independent of the path or process by which the substance is transferred from one state to the other. In a refrigerating system the refrigerant is a working substance used for the purpose of absorbing heat from a *source* (the evaporator) and discharging it through a *sink* (the condenser). Some type of heat pump must be located between the source and sink so that the energy absorbed by the refrigerant at the evaporator may be transferred to the condenser for discharge. In order to predict or study the performance of a refrigerating system it is necessary that the refrigerant be defined throughout the various points of the system in terms of its measurable properties.

For most engineering purposes, including the field of refrigeration, the seven most essential properties¹ are the following:

- (1) Pressure, pounds per square inch absolute (psia)
- (2) Volume, cubic feet (V)
- (3) Temperature, F or absolute
- (4) Kinetic energy, foot-pounds
- (5) Internal energy, Btu
- (6) Enthalpy, Btu (H)
- (7) Entropy, Btu per F (S)

The first three of these properties need no further discussion, since they are readily comprehensible and may be determined by direct observation of instruments or by simple measurements. They are used either in their absolute values or in differential. The remaining four are more complex in their conception and are further discussed in the following sections. The last three of these are used almost entirely as differences, and the numerical values are referred to arbitrary datum levels chosen for their simplification of problems.

A working substance may exist in any of three states or phases: *solid*, *liquid*, or *gas*. The gaseous phase may be further divided for engineering purposes into two regions: the *region of vapors* and the *region of true gases*. The division, as noted previously, is a purely arbitrary one, vaguely defined by the degree of conformity to the perfect gas laws. In most refrigerating systems the refrigerant alternates between the liquid and vapor phases. In the little-used air refrigerating machine the working substance is confined to the gaseous phase. In the manufacture of dry ice or solid carbon dioxide, the solid, liquid, and gaseous phases are all encountered.

3.4. Kinetic Energy. A body in motion will do work upon other objects if its velocity is decreased but must have work done upon it by an outside source if its velocity is to be increased. The energy that a body possesses by virtue of its motion is termed *kinetic energy* and is defined as

¹ See Appendix Table A.1 for corresponding letter symbols.

$$\text{Kinetic energy} = \frac{WV^2}{2g} \quad (3.3)$$

and the work done by a body during a change of velocity is

$$w_k = \frac{W}{2g} (V_1^2 - V_2^2) \quad (3.4)$$

where the subscripts 1 and 2 refer to the initial and final conditions, and W denotes weight, g gravitational acceleration and V velocity. In refrigeration this property of the working medium is useful in analyzing the action of steam-jet units, centrifugal compressors, and refrigerant injector nozzles, and in studying the throttling process. In most cases, however, the energy interchange involved during a velocity change of the working substance is sufficiently small to be negligible.

3.5. Internal Energy. The *internal energy* of a substance as applied to refrigeration is the kinetic energy of its molecules plus the potential energy possessed by virtue of the molecular arrangements assumed against mutually attractive forces. Subatomic internal energy, sometimes included by the physicist and chemist, has no common application in engineering thermodynamics, and atomic energy, at present, is considered only in engineering processes involving combustion.

The *internal kinetic energy* of a substance is commonly termed *sensible heat*, since it increases in relation to the absolute temperature of the body. The *internal potential energy* of a body is commonly termed *latent heat*, or *heat of disgregation*, as it is evidenced only during changes of phase and therefore is unaccompanied by any changes in temperature. Thus, equivalent weights of water, ice, and steam at 32 F would all possess the same internal kinetic energy, but the internal potential energy, and therefore the total internal energy itself, would be different because of the difference in phase. The internal energy of an ideal gas would be equal to the sensible heat alone, because its concept inherently allows no change of phase.

3.6. Enthalpy. *Enthalpy* is an arbitrary composite energy term defined as the sum of the internal energy of a substance plus the product of the absolute pressure and the volume in heat units. Expressed mathematically, this is

$$h = u + \frac{pv}{J} \quad (3.5)$$

Although enthalpy is one of the most practical and widely used thermodynamic energy terms, it is probably one of the least thoroughly understood. Under all conditions the internal energy portion of the term consists of stored energy and is a property of the substance. The pv/J product is also a property of the substance but does not indicate stored energy. There is, however, a form of energy indicated by these same factors but present only during steady, continuous flow of the substance.

In the case of a flowing fluid such as a refrigerant, the pv/J term represents the energy necessary to maintain steady flow. If a pound of fluid having a volume v is made to flow through a pipe of cross-sectional area A by a pressure p , the energy required will be the product of the acting force F times the distance through which it acts l , or Fl . But, since $F = pA$ and $v = Al$, $Fl = pAl = pv$. Expressed in Btu per pound of substance, this energy, sometimes termed flow work, is therefore pv/J , the same expression as the term in enthalpy. As energy it exists only during flow. Properties of a substance, like enthalpy, are always present under either flow or static conditions.

Because refrigerant gases are usually in the vapor region, and therefore the perfect gas laws do not apply, the concept of enthalpy, is widely used in refrigeration engineering to indicate energy exchanges. The change in enthalpy is a measure of heat transferred in a constant-pressure process. This is a common process in refrigeration, and refrigeration cycles are usually plotted on *pressure-enthalpy* or *Mollier* coordinates. Enthalpy values are included in all tables of refrigerant properties. The datum of zero enthalpy for refrigerants is usually chosen as -40°F , both because of the convenient size of the resulting numerical values and also because, until recent years, refrigeration below -40°F was comparatively rare. Low-temperature refrigeration applications necessitate the introduction of negative enthalpy values.

3.7. Entropy. *Entropy*, like enthalpy, is a mathematical function of the observable properties of a substance, and therefore changes in entropy are not evident to the human senses. An increase in entropy indicates an increase in the degradation of energy or a decrease in the availability of energy. Mathematically, entropy changes between two states, 1 and 2, are defined as

$${}_1ds_2 = \int_1^2 \frac{dQ}{T} \quad (3.6)$$

where Q denotes quantity of heat, T absolute temperature, and s entropy. This expression occurs repeatedly in thermodynamic analyses, and entropy is a useful coordinate in plotting processes and making charts. Differences in entropy and not absolute values are used; and in the case of refrigerants, as with enthalpy, the zero datum is usually chosen as -40°F .

A common hydraulic analogy to entropy is found in the potential energy of position available at the top of a waterfall and the unharnessed dissipation of that energy in its drop to the foot of the falls. The potential energy of the water is first converted into kinetic energy during free fall and is then dissipated as heat upon impact. The amount of energy remains unchanged, but it has suffered a degradation and is no longer easily converted into useful work as it might have been had a water

wheel or turbine been interposed in the path of its fall. This degradation is loosely equivalent to the throttling of a gas from a high pressure to a low pressure through an orifice in that again there is a decrease in the availability of the energy although the energy content itself remains the same. In this case the energy degradation is measurable as an increase in entropy and has not been accompanied by the accomplishment of useful work. The interposing of an expansion engine in lieu of the orifice would have accomplished conversion of part of the energy to useful work.

3.8. Reversibility. A *reversible process* is one that is completely controlled and thus yields a maximum amount of useful work. It can be made to traverse in the reverse order all the stages through which it has progressed. Such a process is characterized by a continuous series of equilibrium states each of which can be defined numerically and thus plotted graphically. No actual process is completely reversible, but many engineering processes approach reversibility and equilibrium to an extent sufficient to warrant their analysis as completely reversible.

In refrigeration systems the processes of compression, condensation, and evaporation all approach reversibility, and each stage of these processes may be represented graphically with reasonable accuracy. The process of throttling the refrigerant through an expansion valve, however, is completely irreversible and is characterized as entirely turbulent and incapable of proceeding in the reverse order. Here only the initial and final states may be represented truly by graphical means. Similarly the transfer of heat from the condenser, from the compressor, and to the evaporator to or from their surrounding media are all irreversible processes. Thus with a refrigerating system, as with all other actual systems, specific portions of the entire process approach reversibility, whereas others are irreversible. Obviously, if any portion of the process as a whole is irreversible, the refrigeration process itself must also be irreversible.

3.9. Graphical Representation. Any thermodynamic cycle consisting of a series of processes such as the vapor-compression refrigeration cycle may be represented graphically by plotting the processes on coordinates that represent properties of the working substance. Of the seven properties listed in §3.2, two may be eliminated from consideration as practical coordinates for refrigeration analyses. Internal energy does not present as complete a measure of energy conditions as does enthalpy and is therefore a less desirable coordinate; kinetic energy is of minor consideration in most refrigeration processes and therefore may also be discarded. With the five remaining properties there are 10 possible coordinate combinations, and of these the p - v , T - s , and p - h are most commonly used.

The p - v diagram is useful in analyzing compressor operation, since the area under a curve representing a reversible process on these coor-

dinates is equivalent to the work done on the gas. For a nonflow reversible process proceeding between two state points the work is $\int p \, dv$, but for a fluid compressed under flow conditions the initial and final flow energy must also be considered, and the resulting work is equivalent to $\int v \, dp$.

The T - s diagram is useful under representation of the entire refrigeration process because the area under a reversible-process curve on these coordinates is equivalent to the heat gained or lost, $Q = \int T \, ds$.

In refrigeration, as in many other engineering fields, the solution of practical problems is sometimes facilitated if coordinates are chosen which, instead of possessing areas having common physical significance, form a graphical substitute for tables of properties. The Mollier p - h chart is widely used for this purpose in refrigeration and is acceptable whenever great precision is not necessary. From this chart the observable properties of pressure, volume, and temperature, and the functional properties of enthalpy and entropy may be obtained directly. The internal energy may be calculated from the relationship

$$h = u + \frac{pv}{J} \quad (3.5)$$

Mollier p - h charts for several of the common refrigerants are available in the envelope in the back of this book.

3.10. Thermodynamic Laws. There are two major premises upon which the science of thermodynamics is based. Because no exceptions to these rules have ever been experienced, either accidentally or through controlled experiments, and because all attempts to disprove them have failed, they are termed thermodynamic laws and have been used as a foundation for further developments.

The *first law of thermodynamics* states that heat and mechanical energy are interconvertible and neither can be created nor destroyed. This is a limited form of the more general *law of conservation of energy*, which states that all forms of energy are interconvertible and can neither be created nor destroyed. Although these statements are all-inclusive, no inference can be drawn that the conversion of one energy form to another is necessarily complete. Only a portion of a given definite amount of heat energy can be converted to mechanical or to electrical energy, whereas all energy forms can be converted completely into heat energy. This ability for complete conversion into heat accompanied by a general tendency for all energy to be dissipated eventually in this manner has led sometimes to the description of heat as a "low-grade energy." For example, the power input to an electric motor operating a refrigeration compressor is all converted, eventually, into heat and is absorbed as heat through the condenser or lost to the surroundings by convection, conduction, and radiation. All electric-motor energy losses, such as resistance, windage, hysteresis, and friction, are transformed into heat. All electric power converted into mechanical energy by the motor is

supplied to the compressor for compression of the refrigerant. This mechanical energy is eventually extracted at the condenser as heat, and all frictional losses are dissipated as heat. Even the energy transformed into sound in the operation of the plant is so dissipated.

The *second law of thermodynamics*, according to Clausius, states that it is impossible for a self-acting machine unaided by any external agency to convert heat from one body to another at higher temperature. Other statements of this same premise are that heat will not of itself flow from one body to another body maintained at a higher temperature, and that no machine, actual or ideal, can both completely and continuously transform heat into mechanical energy. All these statements imply the same principle, but Clausius' is probably the most closely allied to refrigeration. The second law enables direct limitations to be placed upon the theoretical maximum operating efficiency that can be attained by either a compression or an absorption refrigerating machine operating under any specified set of conditions.

The *Nernst heat theorem*, frequently referred to as the *third law of thermodynamics*, states that it is impossible by any procedure, no matter how idealized, to reduce any system to absolute zero of temperature in a finite number of operations. Its engineering applications are few, and its discussion is beyond the scope of this text. Some of its corollaries, however, indicate interesting phenomena at extremely low temperatures. The Nernst heat theorem may be interpreted to indicate that the coefficient of volume expansion and the specific heat of a fluid approach zero as the temperature approaches zero. Such conclusions are in accordance with Grüneisen's² investigations, which indicate an increase in the thermal coefficient of expansion for most pure metals as the temperature increases. In addition, his experiments show that the ratio of the coefficient of linear expansion of a substance to its specific heat at constant pressure approaches a constant at temperatures down to at least -170°C .

3.11. Energy Equations. It is frequently necessary to evaluate the energy interchanges occurring to or from a working medium during a process. The development of the quantitative relationships necessary to accomplish such an evaluation is based upon five basic equations, known as the *specific heat equation*, the *nonflow* and the *steady-flow work equations* for an expanding substance, the *nonflow* or *simple energy equation*, and the *steady-flow* or *general energy equation*.

The specific heat equation, based upon the definition of the specific heat of a substance for a given process, enables determination of the heat-energy interchange over a known temperature change. The general expression for this energy relationship is

$${}_1Q_2 = W \int_{t_1}^{t_2} c \, dt \quad (3.7)$$

² Grüneisen, *Ann. Physik*, Vol. 26 (1908), p. 211.

where c is the specific heat and t the temperature. This reduces to

$${}_1Q_2 = Wc(t_2 - t_1) \quad (3.8)$$

if the temperature range is sufficiently small to allow the specific heat to be considered as a constant. Care should be taken that these equations are not employed over a discontinuity of the specific heat such as occurs in refrigeration systems during a change of state of the refrigerant.

When there is no transfer of the working substance during a process, it is termed nonflow; when there is a continuous and steady flow of the working medium, it is termed steady flow. Processes involving inter-

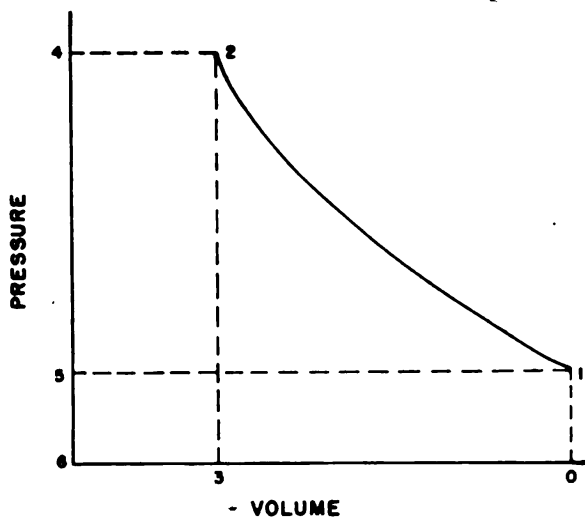


Fig. 3.2. Compression process.

mittent flow with rapid cycling, such as the compression of a refrigerant, are usually treated as steady flow, although, if desired, the individual parts of the cycle may be treated as nonflow. The nonflow work equation for an expanding substance may be derived from the energy required to move a piston of known area through a known distance against a known force and is equal to

$${}_1W_2 = \int p \, dv \quad (3.9)$$

In the compression process represented in Fig. 3.2 this is equivalent to the area under curve 1-2 (area 01230). If during compression, however, steady-flow conditions exist, this area represents only a portion of the work required for compression. The working medium enters the compression process possessing flow energy p_1v_1 (area 01560) and is discharged with flow energy p_2v_2 (area 24632). The difference in these flow energies added to the expression of equation 3.9 results in the steady-flow work equation equivalent to the process

$${}_1W_2 = \int v \, dp \quad (3.10)$$

The *steady-flow* or *general energy equation* is a mathematical statement of the energy conservation law including all flow process terms normally

encountered in heat engineering. It is merely an energy balance equating all energy entering a closed system to that leaving the system. The only restrictions on its application are that there must be continuous, constant flow of the working medium with no reservoirs capable of energy storage or rejection. The expression for the steady flow energy equation is

$$\frac{V_1^2}{2gJ} + u_1 + \frac{p_1 v_1}{J} + E_{P_1} + Q = \frac{V_2^2}{2gJ} + u_2 + \frac{p_2 v_2}{J} + E_{P_2} + \frac{W_k}{J} \quad (3.11)$$

where u is the internal energy per unit weight and E_p the potential energy. All energy terms in the equation are associated with the working

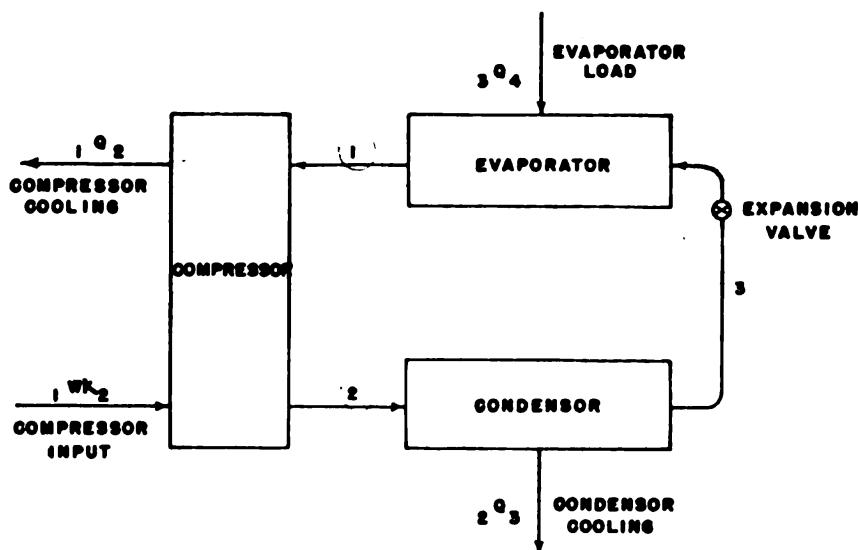


Fig. 3.3. Vapor-compression refrigeration system.

medium, with the exception of work and heat, which are added or subtracted from the system independently.

The *nonflow* or *simple energy equation* is a simplified mathematical statement of the general energy equation eliminating all terms not involved in a nonflow process as well as those which are usually negligible. Its expression is

$$dQ = dU + \frac{dW_k}{J} \quad (3.12)$$

where U is the total internal energy.

For some processes the equation is even further simplified by the elimination of an additional term. Thus during a constant volume process the term dW_k/J is zero, during an isothermal process the term dU is zero, and during an isentropic process the term dQ is zero.

The vapor-compression refrigeration system shown in Fig. 3.3 may be subjected to analysis by the general energy equation, with each unit of the system considered separately or the entire system considered as a

whole. If all unnecessary terms are eliminated and radiation and piping losses are disregarded, the following values are obtained:

For the compressor:	$Wh_1 - \frac{1W_{k_2}}{J} = Wh_2 - {}_1Q_2$
For the condenser:	$Wh_2 = Wh_3 - {}_2Q_3$
For the evaporator:	$Wh_3 - {}_3Q_1 = Wh_1$
For the complete system (adding component equations algebraically)	${}_3Q_1 + \frac{1W_{k_2}}{J} = {}_1Q_2 + {}_2Q_3$

Thus the heat absorbed through the evaporator plus the work of compression is equal to the heat rejected in cooling the compressor and condenser. In this example the expansion valve has been considered a part of the evaporator, since an energy balance across this valve, assuming the entering and leaving velocities to be equal, would reduce to $h_1 = h_2$.

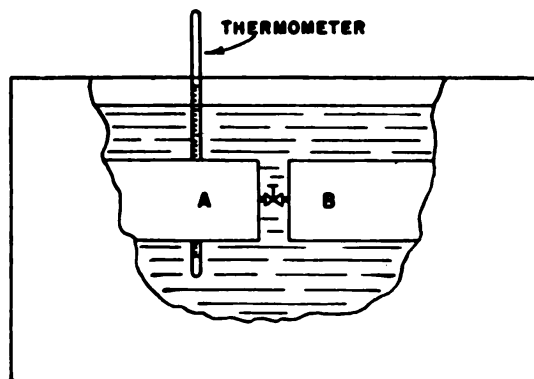


Fig. 3.4. Joule's free-expansion experiment.

3.12. Joule's Experiment and the Joule-Thomson Effect. In 1850 J. P. Joule performed his classical experiment on gases expanding with zero work and without gain or loss of heat. His apparatus, shown diagrammatically in Fig. 3.4, consisted of two chambers A and B connected by a tube and stopcock, all immersed in a water bath. One chamber was evacuated and the other filled with a gas compressed to approximately 22 atmospheres. After sufficient time had been allowed to attain temperature equilibrium, the interconnecting stopcock was opened. Although the gas was allowed to expand to approximately twice its initial volume, the volume of the flask system remained unchanged and hence no external work was done. Observations of the temperature of the surrounding water bath indicated no change after expansion, and therefore there could have been no transfer of heat from the gas to the water. There could also have been no change of internal energy, since, from the energy equation,

$$dU = \frac{dW_k}{J} - dQ = 0$$

The significant fact of this experiment was that both the internal energy and the temperature of the gas remained constant regardless of the changes brought about in the system. It was concluded that the internal energy of a gas is a function of its temperature alone.

Although Joule's early experiments indicated the internal energy of a gas to be a function of temperature alone, his later experiments in 1852 in collaboration with Sir William Thomson (later Lord Kelvin) proved Joule's original apparatus to have been of insufficient sensitivity and his conclusions true only for an ideal gas. In the Joule-Thomson apparatus,

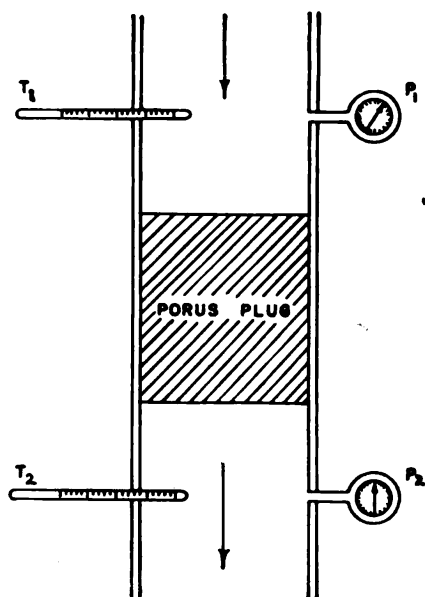


Fig. 3.5. Joule-Thomson porous-plug experiment.

shown diagrammatically in Fig. 3.5, gases were allowed to expand adiabatically under steady-flow conditions through a thermally insulated porous plug of cotton wool. A constant pressure differential was maintained across the plug during flow of the gas, and the entering and leaving temperatures were determined. These and more recent experiments have shown that during adiabatic throttling of a real gas there is a change in temperature with drop in pressure termed the *Joule-Thomson coefficient*.

$$\left(\frac{\partial T}{\partial p}\right)_H = \mu \quad (3.13)$$

This ratio is not a constant but varies with both temperature and pressure, and for each gas there is an *inversion point* (which varies somewhat with pressure) at which the coefficient is zero and the temperature remains constant. Below this inversion point the temperature drops upon throttling and above it the temperature rises. With the exception of hydrogen, which has an inversion temperature of approximately -108°F , practically all common gases will produce cooling upon expansion in the pressure ranges usually encountered.

The Joule-Thomson effect has several practical applications in the field of refrigeration. It has been widely used in the study of the properties of vapors and gases because it is a measure of the departure of an actual gas from the ideal gas. It has also been used as a means of refrigeration to attain low temperatures, for example in the Linde or Hampson systems for liquefaction of gases.

3.13. Thermodynamic Relationships for Gases. By far the majority of all refrigeration processes involve working media that alternate between the vapor and the liquid phases and therefore cannot be treated thermodynamically as gases (§3.14). Since there are some exceptions,

however, such as the air-refrigerating machine, the thermodynamic relationships commonly required for gases³ are summarized for reference in Table 3.1. These relationships may all be derived from *Charles' law*.

$$\frac{V_2}{V_1} = \frac{T_2}{T_1} \quad (3.14)$$

Boyle's law,

$$p_1 V_1 = p_2 V_2 \quad (3.15)$$

the *characteristic equation of gases*,

$$pv = RT \quad (3.1)$$

the *polytropic pressure-volume equation*,

$$pV^n = C \quad (3.16)$$

the *energy equations* (§3.10), and the definitions of *specific heat*, *entropy*, and *enthalpy*.

The paths by which thermodynamic changes occur may be studied best by maintaining one of the properties of the working substance constant. In the *isobaric* process ($n = 0$) the pressure is kept constant, and, therefore, the work involved during steady flow is zero. In the *isometric* process ($n = \infty$) the volume is constant, and the work during a nonflow process is zero. In an *isothermal* process ($n = 1$) the temperature is constant, and there is no change in the internal energy. In the *isentropic* or *reversible adiabatic* process ($n = \gamma = c_p/c_v$) there is, by definition, no rejection or addition of heat, and consequently there is also no change in entropy.

In portions of many practical cycles, however, it is impossible to reproduce these theoretical processes, and the actual thermodynamic path between state points can then be defined only by the polytropic process in which n may assume any value. In addition a working medium may expand by uncontrolled or free expansion (irreversible adiabatic process). During such a process no work is accomplished, no heat is added to or rejected from the working medium, and there is no change in internal energy or enthalpy (assuming the initial and final velocities of the expanding substance are constant). There is, however, an increase in entropy, indicating a decrease in availability of energy. In the vapor refrigeration cycle, throttling of the refrigerant through the expansion valve is an irreversible adiabatic process.

3.14. Thermodynamic Relationships for Vapors. Most refrigerants when operating in the gaseous phase are sufficiently close to the saturation curve to be classed as vapors, and therefore the gas laws no longer apply.

³ For a complete discussion and the derivation of these relationships, reference should be made to any standard thermodynamics text.

TABLE 3.1
THERMODYNAMIC RELATIONSHIPS FOR GASES*
(Constant specific heat)

Process	Isometric or Constant Volume	Isobaric or Constant Pressure	Isothermal or Constant Temperature	Isentropic or Constant Entropy (Reversible Adiabatic)	Polytropic	Free Expansion (Irreversible Adiabatic)
Pressure-volume relationship $pV^n = C$	$n = \infty$ $V_1 = V_2$	$n = 0$ $p_1 = p_2$	$n = 1$ $p_1 V_1 = p_2 V_2$	$n = \gamma$ $p_1 V_1^\gamma = p_2 V_2^\gamma$	$p_1 V_1^n = p_2 V_2^n$	
Heat added to gas ${}_1Q_2 =$	$W_{C_v}(t_2 - t_1)$	$W_{C_p}(t_2 - t_1)$	$\frac{WR}{J} \log_e \frac{V_2}{V_1}$	0	$W \left(\frac{n - \gamma}{n - 1} \right) C_v(t_2 - t_1)$	0
Work done by gas (non-flow process) ${}_1W_2 =$	0	$p(V_2 - V_1)$	$pV \log_e \frac{V_2}{V_1}$	$\frac{p_1 V_1 - p_2 V_2}{\gamma - 1}$	$\frac{p_1 V_1 - p_2 V_2}{n - 1}$	0
Work done by gas (steady-flow process) ${}_1W_2 =$	$V(p_2 - p_1)$	0	$pV \log_e \frac{p_2}{p_1}$	$\frac{\gamma}{\gamma - 1} (p_1 V_1 - p_2 V_2)$	$\frac{n}{n - 1} (p_1 V_1 - p_2 V_2)$	0
Gain in entropy of gas ${}_1S_2 =$	$W_{C_v} \log_e \frac{T_2}{T_1}$	$W_{C_p} \log_e \frac{T_2}{T_1}$	$\frac{WR}{J} \log_e \frac{V_2}{V_1}$	0	$W \left(\frac{n - \gamma}{n - 1} \right) C_v \log_e \frac{T_2}{T_1}$	$\frac{WR}{J} \log_e \frac{V_2}{V_1}$
Gain in internal energy of gas ${}_1U_2 =$	$W_{C_v}(t_2 - t_1)$	$W_{C_p}(t_2 - t_1)$	0	$-W_{C_v}(t_2 - t_1)$	$W_{C_v}(t_2 - t_1)$	0

* Symbols are defined in Appendix Table A.1.

Most refrigerants also operate in the liquid phase, infrequently even in the solid phase, and may coexist in varying proportions in two phases at the same time. Although this greatly complicates the thermodynamic relationships, the actual solution of refrigeration problems is simplified, since process equations are not feasible, and the use of graphical or tabular presentations of the thermodynamic properties is necessary.

The states in which a refrigerant or similar substance may exist and the conditions under which transition between states may occur is best studied by reference to diagrams. Figure 3.6(a) is a *pressure-volume* diagram, 3.6(b) a *temperature-entropy* diagram, and 3.6(c) a *pressure-enthalpy* diagram, all for a typical refrigerant that contracts upon freezing. On all these diagrams the boundary curves separating the phase areas are indicated by identical numbering. The *saturated-liquid line* 3-4 and the *saturated-vapor line* 4-6 (commonly referred to as the *saturation line*), together with the *triple-point line* 2-3-5, bound the area in which the liquid and vapor phases coexist in varying proportions. In this area the weight proportionality of the nonhomogeneous mixture of the two phases is termed *quality*; a quality of 0.90 would indicate a mixture of 10 per cent saturated liquid and 90 per cent saturated vapor.

In the area to the left of the *saturated-liquid line* and above the *triple-point temperature* the substance is a *subcooled liquid*. In the area to the right of the *saturated-vapor line*, both above and below the *triple-point temperature*, the substance is a *superheated vapor* or *gas*.

The *critical point* 4, at the junction of the *saturated-liquid* and *saturated-vapor* lines, defines the *critical temperature* above which liquefaction of the substance cannot occur.⁴ Above the *critical pressure* the

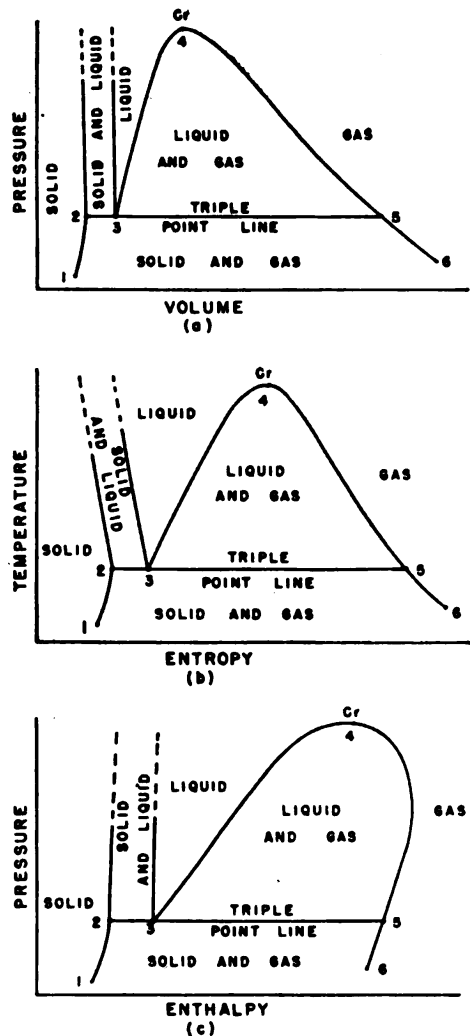


Fig. 3.6. Typical refrigerant diagrams.

⁴ P. W. Bridgman and Gustav Tammann have shown that under extremely high pressures, in the order of a half million pounds per square inch, liquefaction can still occur above the critical temperature.

latent heat of vaporization becomes zero, the defining boundary line between the liquid and gaseous phases disappears, and there are no phenomena such as condensation and vaporization. There is no conclusive evidence of any change whatsoever when the refrigerant passes from the liquid to the vapor state, and hence above the critical point these regions merge.

The *saturated-solid line* 1-2, the lower portion of the saturated-vapor line 5-6, and the triple-point isotherm 2-3-5 bound the area in which the solid and vapor phases coexist in varying proportions. In the area to the left of the saturated-solid line and below the triple-point temperature the substance is a subcooled solid. The *triple point* 2-3 (at triple-point temperature, t_i) is a unique series of state points at which the substance may exist in all phases—solid, liquid and gas—in equilibrium. Below the triple-point temperature the heat required to change the substance from a solid directly to a gas is termed *latent heat of sublimation*. At the triple-point temperature the heat required to change from a solid to a liquid (along 2-3) is termed *latent heat of fusion*. Above this temperature the heat required to change from a liquid to a vapor is termed *latent heat of vaporization*.

3.15. Properties of Vapors. Tables of the properties of various refrigerants are presented in the Appendix and are similar in both form and application to standard steam tables. The saturated-refrigerant values are tabulated for temperature, pressure, specific volume, enthalpy, and entropy for both the saturated-liquid and saturated-vapor states. In the superheated-refrigerant tables the values of temperature, pressure, specific volume, enthalpy, and entropy are included. Except where noted otherwise, the enthalpy and entropy values have been computed from a -40°F base instead of the 32°F base used for water vapor or steam.

As in the case of similar working mediums capable of existing in vapor and liquid phases, the properties of a refrigerant state point located between the saturation curves may be calculated from the tabulated values if the quality is known. The enthalpy of such a state point representing two coexistent phases would be the enthalpy of the liquid plus the product of the quality and the latent heat of vaporization. Thus

$$h_x = h_f + xh_{fg} \quad (3.17)$$

Similarly, the entropy would be

$$s_x = s_f + xs_{fg} \quad (3.18)$$

and the specific volume

$$v_x = v_f + xv_{fg} \quad (3.19)$$

Internal energy values are not usually tabulated but may be calculated from equation 3.5.

3.16. Processes. In the vapor-compression refrigeration cycle both condensation and evaporation take place at constant pressure. By differentiating the defining equation for enthalpy

$$dh = du + \frac{p}{J} dv + \frac{v}{J} dp$$

it is found that

$$dh = dQ$$

as the term $v dp/J$ drops out for constant pressure, and $dQ = du + p dv/J$ is the simple energy equation 3.11 for a nonflow process.

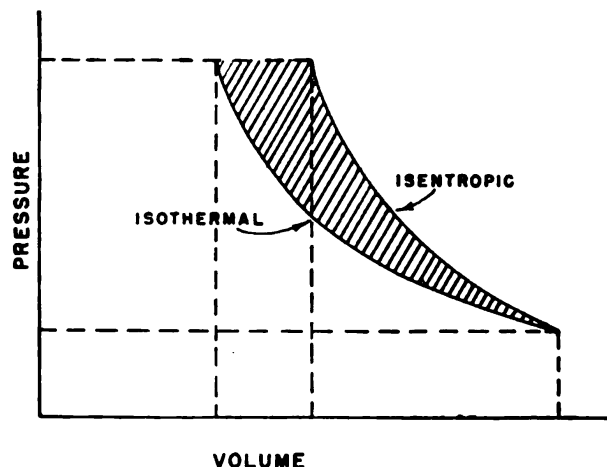


Fig. 3.7. Isothermal vs. isentropic compression.

For a throttling process such as occurs through an expansion valve, the terms involved in the general energy equation 3.12 may be written between the initial and final states, as follows:

$$\frac{V_1^2}{2gJ} + u_1 + \frac{p_1 v_1}{J} = \frac{V_2^2}{2g} + u_2 + \frac{p_2 v_2}{J}$$

By definition, $h = u + pv/J$, and if the velocities of flow are assumed to be equal, then

$$h_1 = h_2$$

If the throttling terminates between the saturation curves,

$$h_1 = h_{f_2} + x_2 h_{fg_2} \quad (3.20)$$

and the quality at the end of expansion is

$$x_2 = \frac{h_1 - h_{f_2}}{h_{fg_2}} \quad (3.21)$$

Since the area on the diagram under an isothermal compression curve is greater than that under an isentropic compression curve as shown in Fig. 3.7, it would at first appear that the isentropic process would be

desirable from the standpoint of reduced work of compression. However, this is true only for a nonflow process; for a steady-flow process the reverse is true (assuming equal velocities and hence equal kinetic energy terms), since the flow work imparted at discharge from an isothermal compression is much less than that for isentropic compression. The work required during compression for a steady-flow process is equivalent to the area between the curve and the pressure axis. Actually, isothermal compression is difficult if not impossible to approach in practice, because the refrigerant gas enters the compressor at a much lower temperature than the cylinder walls. This temperature difference, together with an internal energy increase resulting from compression, tends to prevent an approach to the isothermal.

The compression process of the vapor refrigeration cycle approaches the isentropic more closely than any other. Principally because of throttling through the compressor valves and because of absorption and rejection of heat by the cylinder walls, there are some deviations that depend upon the compressor design and the refrigerant. The theoretical vapor-compression refrigeration cycle, however, is assumed to include isentropic compression.

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PROBLEMS

3.1. A refrigerant is compressed from 30 to 180 psia, and the specific volume decreases from 9 to 2 cu ft per pound. Determine the polytropic exponent of compression.

3.2. A refrigerant gas with 10 cu ft per pound specific volume is compressed polytropically from 10 psia to 66 psia. If $n = 1.25$, determine the specific volume at the end of compression.

3.3. One pound of saturated liquid ammonia is expanded through a throttling valve from a condenser pressure of 180.6 psia to an evaporator pressure of 38.51 psia. Determine (a) the refrigerant quality at discharge, (b) the change in specific volume, and (c) the final quality if the refrigerant is subcooled 10 F before expansion.

3.4. A refrigerating system develops 8 tons while drawing 9 kw electrical energy. If 80 per cent of this energy appears as an increase in refrigerant enthalpy, determine the Btu per minute rejected to the condenser cooling water.

3.5. Liquid refrigerant, in flowing from a receiver to an expansion valve, gains 15 ft in elevation and 3 Btu per pound from external sources of heat. If the velocity remains essentially constant, determine the change in enthalpy.

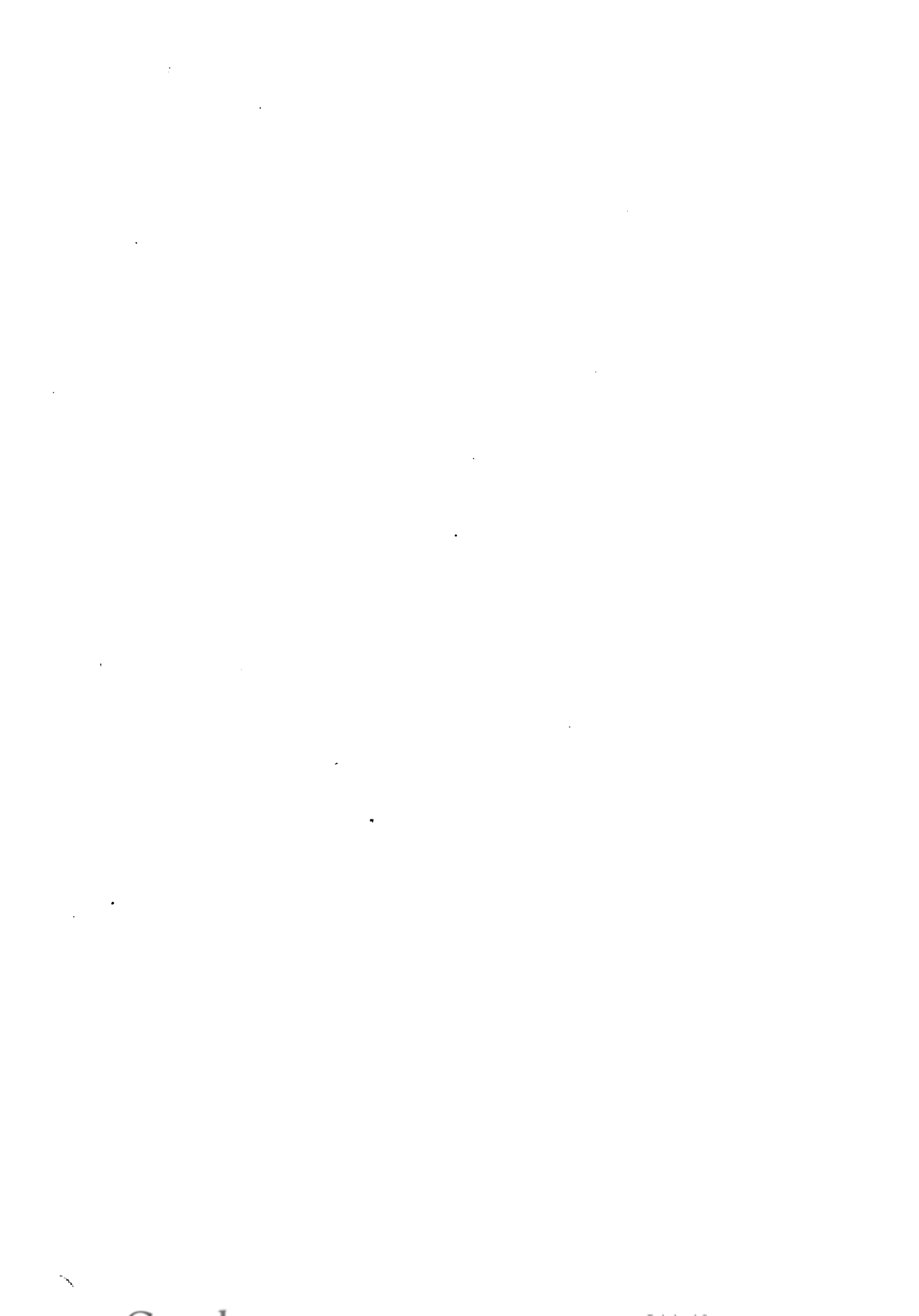
3.6. Establish in general terms an energy balance between liquid refrigerant at comparative rest in a receiver and the refrigerant entering the expansion valve. Assume frictionless tubing.

3.7. Refrigerant vapor is compressed at a rate of 3 pound per minute with an increase in enthalpy of 3.5 Btu per pound. If 0.6 hp is required for compression, determine the Btu per hour rejected to the cylinder cooling water.

3.8. Saturated methyl chloride vapor is compressed from 20 to 100 psia and 130 F. Determine (a) the work of compression in Btu per pound for the process and (b) the work of compression for the steady-flow cycle. See Appendix Table A.4 for properties of methyl chloride.

3.9. From the general energy equation, determine the Btu per pound rejected to the compressor cooling water in Problem 3.8(b).

3.10. If it is assumed that a linear relationship exists between temperature and entropy during compression, determine for Problem 3.8(b) the approximate amount of heat rejected to the cooling water from the area under the compression curve. What is the percentage of error for this approximation?



PART II
Refrigeration Cycle Analysis
and
Compression Refrigeration Machines



CHAPTER 4

Thermodynamics of Vapor Refrigeration

4.1. Introduction. Most practical forms of refrigeration that utilize equipment for the recovery of the refrigerant may be classed broadly as compression systems. Whether this compression is accomplished by reciprocating, rotary, or centrifugal devices, by high-pressure steam ejectors (steam-jet system), or by evaporation from a secondary fluid (absorption system) makes no fundamental difference; all are compression systems basically.

By far the most important systems commercially are those which use a refrigerant alternating between the vapor and the liquid phases. This class¹ also contains the reciprocating, rotary, centrifugal, steam-jet, and absorption systems. These systems, operating cyclically between two pressures and using a two-phase working medium, may, in the broadest terminology, be classed as *vapor-compression systems*. From a practical standpoint, however, such an all-inclusive classification is undesirable. Therefore the absorption and steam-jet systems are, for clarity, almost always so designated, and the term "vapor-compression system" is reserved for reciprocating and rotary systems using a two-phase refrigerant. The centrifugal system, though differing from the reciprocating and rotary units only in that it is not positive in displacement, is usually classed separately for convenience. These clear and commonly used designations will be adhered to throughout this text.

4.2. The Carnot Cycle. The Carnot cycle is an ideal, thermodynamically reversible cycle, first investigated by Sadi Carnot in 1824 as a measure of the maximum possible conversion of heat energy into mechanical energy. In its reversed form it is used as a measure of the maximum performance of refrigeration equipment. Although it cannot be applied in an actual machine because of the impossibility of obtaining a completely reversible engine, it is nevertheless extremely valuable as a criterion of inherent limitations.

The Carnot cycle, shown graphically in Fig. 4.1 on p - V and T - s coordinates, consists of an adiabatic expansion and an isothermal expansion followed by an adiabatic compression and an isothermal compression to form a closed cycle. Since the areas on the T - s diagram represent

¹ One exception not in this group is the air-compression refrigeration cycle, once widely used in marine and naval refrigeration and now reappearing in some aviation applications (see §6.2).

actual heat quantities supplied or rejected, and since all paths are here parallel to the coordinate lines, it is simplest to analyze the cycle on these coordinates. The Carnot cycle when operating as a heat engine will have the efficiency

$$\begin{aligned}\eta &= \frac{\text{net work}}{\text{heat supplied}} = \frac{\text{heat supplied} - \text{heat rejected}}{\text{heat supplied}} \\ &= \frac{(T_1 - T_2)(s_b - s_a)}{T_1(s_b - s_a)} = \frac{T_1 - T_2}{T_1}\end{aligned}\quad (4.1)$$

where T denotes absolute temperature and s entropy.

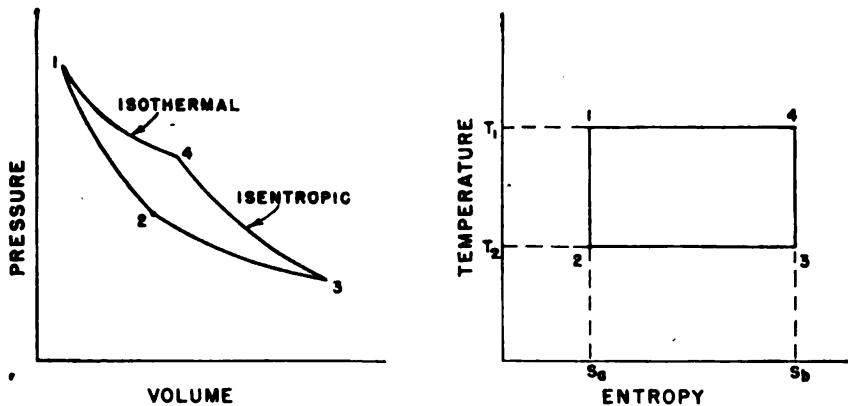


Fig. 4.1. Carnot cycle.

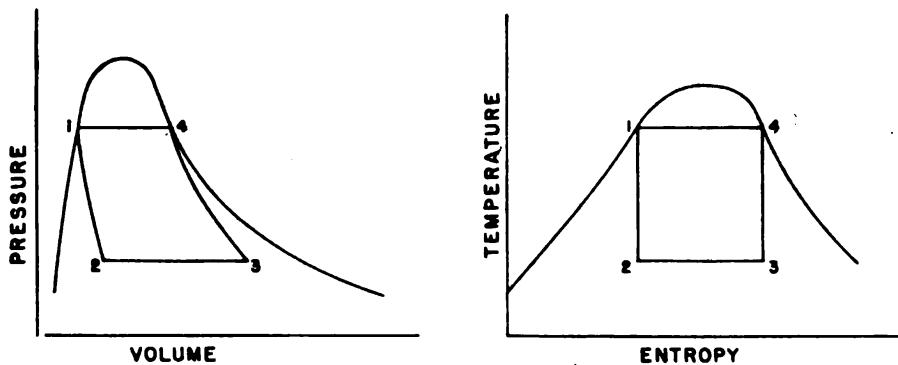


Fig. 4.2. Two-phase Carnot cycle.

Proof that this expression represents the maximum efficiency that can be obtained by any heat engine operating between two temperature levels may be found in any standard text on thermodynamics.

The Carnot cycle shown in Fig. 4.1 is for a working medium operating within the gaseous phase only. It is possible, however, to devise a Carnot cycle operating between the liquid and vapor phases within the saturated region. Such a cycle is shown in Fig. 4.2 on both p - V and T - s coordinates and resembles the Carnot cycle for gases on the T - s diagram. Within the saturated region, however, the isothermal lines are also constant-pressure lines, so that on the p - V diagram these proc-

esses appear accordingly. Such a two-phase Carnot cycle may be approached more closely in practice than a single-phase Carnot cycle because the isothermal processes are closely followed in the saturated region during change of phase.

4.3. Reversed Carnot Cycle and Coefficient of Performance. Since the Carnot cycle is, by hypothesis, a reversible cycle, it is possible to use the reversed Carnot cycle as a measure of the maximum performance to be obtained from a refrigerating machine. In this case shaft work must be applied to the cycle, the working medium is expanded adiabatically from T_1 to T_2 , it absorbs heat isothermally at T_2 with an entropy increase from s_a to s_b , it is compressed adiabatically from T_2 to T_1 , and it discharges heat isothermally at T_1 with an entropy decrease from s_b to s_a (see Fig. 4.1). Thus the Carnot cycle, when operating in one direction or the other, may be used for three purposes: first, for converting heat energy into mechanical energy (when used as a heat engine); second, for using mechanical energy to absorb heat at some undesirable location and to reject it at some unobjectionable one (when used as a refrigerating machine); and third, for using mechanical energy to absorb heat at some ineffective location and to discharge it at a desirable one (when used as a heat pump). The second and third applications are based upon the reversed Carnot cycle and differ only in the result desired.

The term "coefficient of performance" (c.p.) has been devised to measure the effectiveness of refrigerating machines and is usually defined as the ratio of the refrigeration produced in Btu to the net work supplied in Btu. The term can be applied to an actual refrigeration machine or to a theoretical cycle such as the reversed Carnot cycle. Reference to Fig. 4.1 shows that the refrigeration produced in the reversed Carnot cycle is $T_2(s_b - s_a)$ Btu per pound, and the net work supplied is $(T_1 - T_2)(s_b - s_a)$ Btu per pound. Thus the coefficient of performance is

$$\begin{aligned} \text{c.p.} &= \frac{\text{heat absorbed}}{\text{heat equivalent of net work supplied}} \\ &= \frac{T_2(s_b - s_a)}{(T_1 - T_2)(s_b - s_a)} \\ &= \frac{T_2}{T_1 - T_2} \end{aligned} \quad (4.2)$$

Since the term "coefficient of performance" is analagous to the term "efficiency" applied to heat engines, a broader concept of c.p. is sometimes used in defining it as the ratio of the "desired effect" in Btu per pound to the net energy supplied in Btu per pound. For a Carnot-cycle refrigeration machine this is still

$$\text{c.p.} = \frac{T_2}{T_1 - T_2} \quad (4.2)$$

For a Carnot-cycle heat pump in which the desired effect is the heat rejected,

$$\begin{aligned}\text{c.p.} &= \frac{\text{heat rejected}}{\text{net work supplied}} \\ &= \frac{T_1(s_b - s_a)}{(T_1 - T_2)(s_b - s_a)} \\ &= \frac{T_1}{T_1 - T_2}\end{aligned}\quad (4.3)$$

and for a Carnot-cycle heat engine,

$$\begin{aligned}\text{c.p.} &= \frac{\text{net work realized}}{\text{heat supplied}} \\ &= \frac{(T_1 - T_2)(s_b - s_a)}{T_1(s_b - s_a)} = \frac{T_1 - T_2}{T_1}\end{aligned}\quad (4.4)$$

The Carnot-cycle coefficient of performance for both a refrigerating machine and a heat pump increases as the spread between source temperature and sink temperature decreases, whereas the reverse is true for the c.p. or efficiency of a heat engine. If the cycle condition under which T_2 is equal to absolute zero is excluded, the c.p. for either a heat pump or a refrigerating machine is always greater than 1.0, and that for a heat engine is always less than 1.0.

EXAMPLE 4.1. A Carnot-cycle machine operates between the temperature limits of $t_1 = 86^\circ\text{F}$ and $t_2 = 5^\circ\text{F}$. Determine the c.p. when it is operated as (a) a refrigerating machine, (b) a heat pump, and (c) a heat engine.

SOLUTION:

(a) refrigerating machine

$$\begin{aligned}T_1 &= 86 + 460 = 546 \\ T_2 &= 5 + 460 = 465 \\ \text{c.p.} &= \frac{T_2}{T_1 - T_2} = \frac{465}{546 - 465} = 5.74\end{aligned}$$

(b) heat pump

$$\text{c.p.} = \frac{T_1}{T_1 - T_2} = \frac{546}{546 - 465} = 6.74$$

(c) heat engine

$$\text{c.p.} = \frac{T_1 - T_2}{T_1} = \frac{546 - 465}{546} = 0.15$$

In this problem the c.p. for the refrigerating machine is 38.3 times greater than that for the heat engine, whereas that for the heat pump is 1.18 times that for the refrigerating machine. This result is to be expected because the heat engine is converting heat energy into mechanical energy, whereas both the refrigerating machine and the heat pump are merely increasing the energy level of heat absorbed from a low tempera-

ture source. The c.p. for the heat pump is greater than that for the refrigerating machine because all mechanical shaft work required to operate the pump is dissipated as heat and added to the "desired effect," or heat discarded at the high temperature.

4.4. Maximum Coefficient of Performance. That the coefficient of performance for a reversed Carnot cycle (or any other reversible refrigeration cycle) is the maximum possible for a refrigeration machine operating between the same temperature levels may be proved in a manner similar to that used to prove that the efficiency of a Carnot-cycle heat engine is a maximum. In Fig. 4.3(a), H is a machine with a c.p. assumed to be

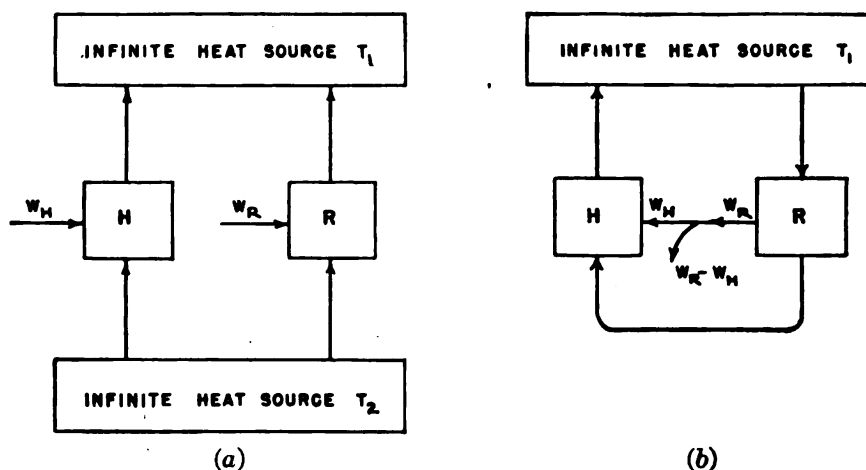


Fig. 4.3.

higher than that for the reversible machine R . Both machines operate as refrigerators between the temperature levels T_1 and T_2 , and both absorb the same amount of heat, Q , at the temperature source T_2 . The shaft energy required by refrigerator H is W_H and by refrigerator R is W_R . The c.p. for refrigerating machine H is Q/W_H , and for machine R is Q/W_R . However, by hypothesis,

$$\frac{Q}{W_H} > \frac{Q}{W_R}$$

and therefore

$$W_R > W_H$$

In Fig. 4.3(b) the reversible machine R has been reversed to operate as a heat engine, and the low-temperature source at T_2 has been replaced by a heat conductor. This procedure is permissible because, again by hypothesis, the amounts of heat flowing between the low-temperature source and either machine are identical. Thus, the system now constitutes a machine which operates from a single source of heat at any temperature above absolute zero. The engine portion of the machine develops sufficient work, W_H , to operate the refrigerator portion plus a surplus of mechanical energy, $W_R - W_H$. If such a machine were possible, work could be produced continuously by any source of heat whatso-

ever, such as the atmosphere, the ocean, or the earth itself. Such a hypothetical machine, known as a *perpetual-motion machine of the second kind*, is impossible to realize because it violates the second law of thermodynamics by continuously and completely transforming heat energy into mechanical energy.

4.5. Theoretical Vapor Compression Cycle. The theoretical vapor-compression refrigeration cycle as actually applied and approached in practice (Rankine form of cycle) is shown in Fig. 4.4(a), (b), and (c) on p - V , T - s , and p - h coordinates. Here 1-2 is the throttling or expansion process, 2-3 is evaporation, 3-4 is compression, and 4'-1 is condensation. Two alternate compression paths, 3-4 and 3'-4' are shown, the former

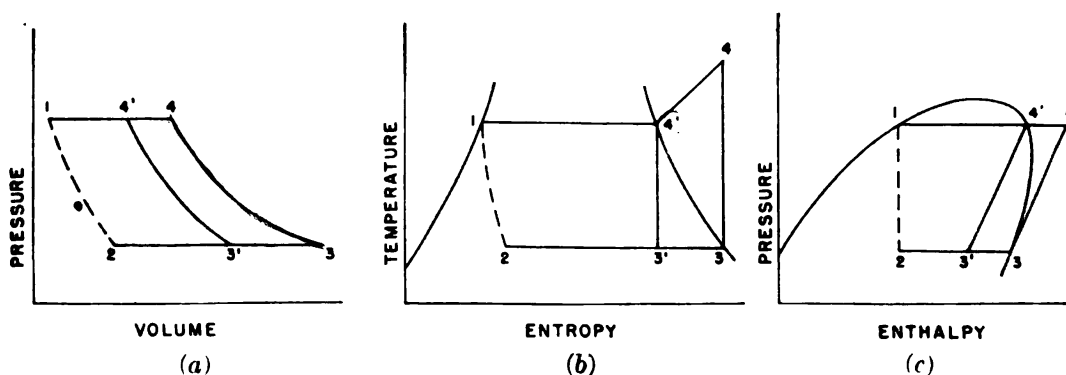


Fig. 4.4. Vapor-compression refrigeration cycle.

termed “dry compression” with the charge entering the compressor initially dry and saturated, and the latter “wet compression” with the charge dry and saturated (theoretically) at the end of compression. Despite the theoretically slightly lower c.p. of dry compression, it is generally utilized in the United States because, among other reasons, there is less danger of damage to the compressor through the entrance of slugs of liquid refrigerant that are not completely vaporized during compression.

No attempt is made to recover the work of expansion in process 1-2, since this positive work—represented in Fig. 4.4(a) by the area between process 1-2 and the pressure-coordinate axis is generally small, and the equipment necessary for its recovery is not economically justifiable. Whereas heat engines are usually large devices centrally located for the production of power in manufacturing plants or central stations, refrigeration machinery is usually limited in application and therefore comparatively low in power input. The recovery of any small amounts of positive work by means of an expansion device or engine is therefore not attempted, and expansion occurs as an irreversible adiabatic process.

Both evaporation (process 2-3' or 2-3) and condensation (process 4'-1) take place within the saturated-vapor region and therefore occur under constant pressure and constant temperature conditions. In the

case of dry compression, however, the gas leaving the compressor is superheated, and the heat of superheat must be removed in the condenser before condensation of the refrigerant can occur. This process 4-4' is one of constant pressure (at condenser pressure) but not of constant temperature.

The compression process (3-4 or 3'-4') in the theoretical cycle is assumed to be isentropic, because this process is more nearly approached in practice. Since the charge entering a refrigerating compressor has been cooled to a temperature greatly below that of the cylinder walls and ports, it is virtually impossible to approach isothermal compression practically. Theoretically it would be desirable if isothermal compression could be assumed, since the work for isentropic compression is greater than that for the isothermal (see Table 3.1). However, from a practical standpoint isothermal compression would have no physical meaning except under conditions of extremely high suction superheat. With normal superheat, isothermal compression would result in the discharge of subcooled liquid refrigerant. This change of state would introduce a considerable reduction in volume and therefore a compression exponent, n , that is less than 1.

It is also evident that if a cooling medium were available that would allow discharge at the suction temperature, there would be no need for the compressor because the cooling medium could be used to replace the refrigerating system.

A comparison of the theoretical vapor compression cycle with the reversed Carnot cycle is shown in Fig. 4.5 on T - s coordinates. This graph indicates the degree of deviation from the reversed Carnot cycle necessary for a theoretical cycle approachable in practice. In making such a comparison, T - s coordinates are of advantage, since the areas shown represent actual heat quantities. In this figure 1-2-3-4-1 represents the Carnot cycle, and 1-2'-3-4'-5-1 represents the vapor compression cycle. The latter deviates from the Carnot cycle in that it includes the superheat "horn" 4-4'-5-4, and expansion follows the irreversible adiabatic 1-2' instead of the isentropic expansion 1-2.

Since the area inside the closed cycle represents the net work added from an outside source, it is evident that the superheat horn increases this required work, whereas the irreversible expansion reduces this work by the area 1-2-2'-1. The heat absorbed through the evaporator is repre-

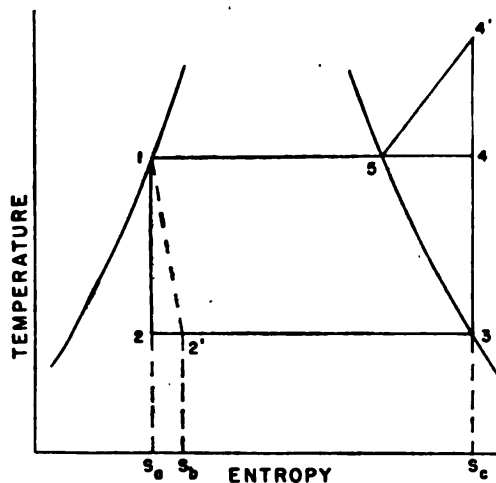


Fig. 4.5. Comparison of reversed Carnot and theoretical vapor-compression cycles.

sented by the area $2-s_a-s_c-3-2$ in the reversed Carnot cycle and by area $2'-s_b-s_c-3-2'$ in the vapor compression cycle. Although the superheat horn has no effect upon the amount of heat absorbed, irreversible expansion reduces the refrigerating effect by the area $2-s_a-s_b-2'-2$. Thus the superheat horn adds to the required net work of the cycle, whereas irreversible expansion decreases this net work somewhat but also reduces the desirable heat absorption in the evaporator by a relatively greater amount.

4.6. Departure from Theoretical Vapor Compression Cycle. The vapor compression cycle as applied in practice differs in several ways from the theoretical cycle shown in Fig. 4.4. Frequently the liquid refrigerant is subcooled before it is allowed to enter the expansion valve, and usually the gas leaving the evaporator is superheated a few degrees before it enters the compressor. This superheating may occur as a result of the type of expansion control used or through a pickup of heat in the suction line between the evaporator and compressor. Compression, although usually assumed to be isentropic, may actually prove to be neither isentropic nor polytropic. In addition, both the compressor suction and discharge valves are actuated by pressure difference, and this process requires the actual suction pressure inside the compressor to be slightly below that of the evaporator and the discharge pressure to be above that of the condenser. Although isentropic compression assumes no transfer of heat between the refrigerant and the cylinder walls, actually the cylinder walls are hotter than the incoming gases from the evaporator and colder than the compressed gases discharged to the condenser. There are numerous other deviations from the theoretical cycle, but most of these are small and consist primarily of heat exchanges, either positive or negative, between parts of the system and the surrounding air. Two additional deviations, not negligible, are the pressure drop in long suction- and liquid-line piping and any vertical differences in head created by locating the evaporator and condenser at different elevations. These two factors are discussed in Chapter 12, Refrigeration Piping.

Figure 4.6 shows the practical vapor compression cycle on T - s coordinates with the sections of the diagram involving entrance of the gas into and discharge from the compressor magnified. In this diagram process 10-11-1-2 represents passage of the refrigerant through the condenser with 10-11 indicating removal of superheat, 11-1 the removal of latent heat, and 1-2 the removal of heat of liquid or subcooling. Process 3-4-5 represents passage of the refrigerant through the evaporator, with 3-4 indicating gain of latent heat of vaporization, and 4-5, the gain of superheat before entrance into the compressor. Both of these processes approach very closely to the constant pressure conditions assumed in theory.

Superheating and Subcooling. The standard rating cycle used for comparison of refrigerating machines and refrigerants under standard

conditions assumes a condenser saturation temperature of 86 F and an evaporator saturation temperature of 5 F. In addition, 9 F liquid subcooling is assumed before the refrigerant enters the expansion valve and 9 F superheating before it enters the compressor. A condensing temperature of 86 F was originally chosen because it corresponds exactly to 30 C, and 5 F was assumed for the evaporating temperature because it corresponds exactly to -15 C. These standards are frequently used in com-

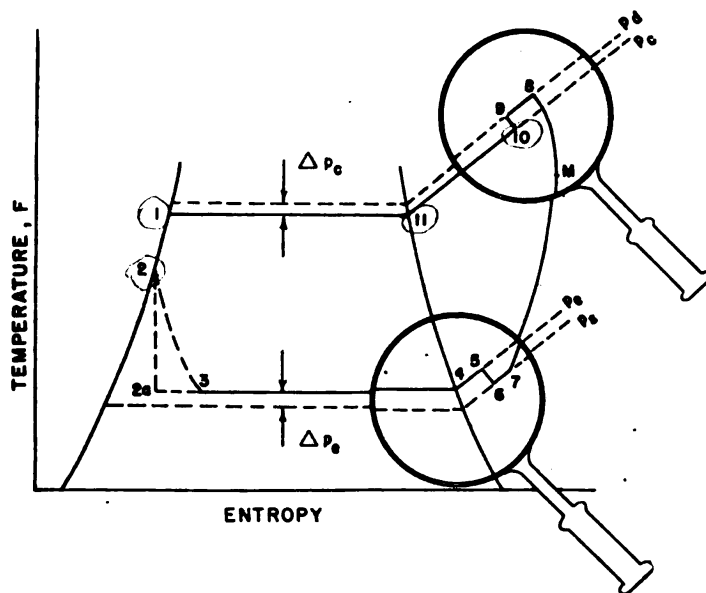


Fig. 4.6. Practical vapor-compression cycle.

paring the ratings of refrigeration equipment. The superheating is advisable in practice because it assures complete vaporization of all the liquid refrigerant before it enters the compressor; moreover, in some instances, variations in superheat temperature serve as a means of modulating the size of opening of the expansion valve (see Chapter 14, Refrigeration Controls). One effect of superheat is to increase the specific volume of the suction vapor and thus also the required compressor displacement per ton of refrigeration developed. A second effect is to increase the enthalpy of the vapor and thus, provided this superheating represents useful refrigeration, also the refrigerating effect for each pound of refrigerant entering the compressor. These two effects tend to counteract each other, and with some refrigerants, such as ammonia, the net effect of superheating is, theoretically, to reduce the capacity of the system. With other refrigerants, such as Freon-12, superheating theoretically increases the capacity of the system.

Liquid subcooling is desirable because it increases the refrigerating effect as well as reduces the volume of gas flashed from the liquid refrigerant in passage through the expansion valve. Any such increase in

refrigerating effect is at the expense only of additional cooling water, and the compressor power requirements remain unchanged.

2. *Throttling.* Process 2-3 represents passage of the refrigerant through the expansion valve, both theoretically and practically an irreversible adiabatic path. The end-state points 2 and 3 are correct, but the locus of intermediate points would probably approach path 2-2a-3 more closely. Here path 2-2a is essentially one of constant entropy, and path 2a-3 is the result of an irreversible decrease in velocity after passage through the valve. Path 2-3 more closely represents the passage of a refrigerant through a porous plug. Actually, no true path can be drawn, since the process is a turbulent one with nonuniform distribution from point to point of any of the properties of the expanding mass.

3. *Suction and Exhaust Pressures.* Four constant-pressure lines are shown in Fig. 4.6, with p_e representing evaporator pressure; p_c , condenser pressure; p_s , suction pressure inside the compressor cylinder after passage through the suction valves; and p_d , discharge pressure inside the compressor before passage through the exhaust valves. Path 5-6-7-8-9-10 represents the passage of the gas from the entrance to discharge of the compressor. Path 5-6 represents the throttling action that occurs during passage through the suction valves, and path 9-10 represents the throttling during passage through the exhaust valves. Both of these actions are accompanied by an entropy increase and a slight drop in temperature.

4. *Compression.* Isentropic compression, assumed theoretically, presupposes that there is no transfer of heat between the refrigerant and the cylinder walls during compression. Actually, the cylinder walls assume a temperature at some point between those of the cold suction gases and the hot exhaust gases, and there is a transfer of heat from the walls to the gases during the first part of compression and a reversal of heat flow during the last part of compression. Moreover, after the cold refrigerant gases pass through the suction valves, the gases undergo, prior to compression, a rise in temperature upon contact with the cylinder walls, and these same gases experience a similar temperature drop after compression and prior to exhaust. These last two heat transfers occur essentially at constant pressure and are indicated by paths 6-7 and 8-9, respectively.

Compression of the refrigerant occurs along path 7-8, which is actually neither isentropic nor polytropic. If it is assumed that the quantity of heat absorbed by the gases during the first part of compression is equal to the quantity of heat rejected by these same gases during the last part of compression, then the form of compression curve shown in Fig. 4.6 will be approached. Here the quantity of heat absorbed by the gases is equal to the area under curve 6-7- M , and this quantity must be equal to the quantity of heat rejected as represented by the area under curve M -8-9. Since the temperature at which rejection of heat from the gases takes place is higher than that at which absorption of heat occurs, and since

the quantities of heat absorbed and rejected are equal, there must be a net entropy increase.

If the cylinder walls are water-jacketed and the temperature of the water is below the mean cylinder-wall temperature, there will be an absorption of some of the heat resulting from compression, a decrease of the cylinder-wall mean temperature, and a decrease in the temperature of the discharged gases. These changes will result in a shifting of curve 6-7-*M*-8-9 such that a greater amount of heat will be rejected to the cylinder walls than is absorbed from them, and there will be a smaller increase or even a decrease in entropy. The magnitude of these changes will of

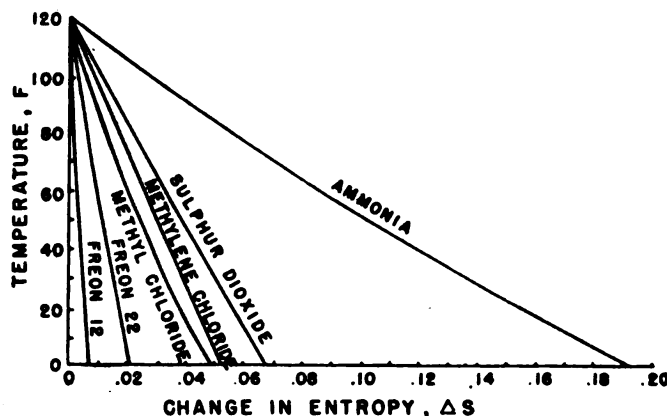


Fig. 4.7. Saturated vapor curves for several common refrigerants.

course depend upon the temperature of the cooling water and the effectiveness of the jacketing. These shifts will result in a reduction of power requirements and an increase in volumetric efficiency (see §4.6). The greatest advantage will be found when refrigerants resulting in high discharge temperatures are used. Thus ammonia compressors are usually supplied with water jackets, whereas Freon-12 compressors are usually only air-cooled.

5. Superheat Horn. The superheat horn 4-4'-5 shown in Fig. 4.5, and its counterpart in Fig. 4.6, represent additional energy of compression required over that for the ideal reversed Carnot refrigeration cycle. The magnitude of this area is dependent upon both the value of n and the slope of the saturated vapor curve for the refrigerant. If compression of a saturated refrigerant follows the vapor curve from the suction to the discharge pressure, there obviously would be no superheat; the degree of departure determines the magnitude of this additional work.

In general the more nearly vertical the vapor saturation line, the smaller is the superheat horn; this statement is correct in all cases if compression is assumed to be isentropic. Figure 4.7 shows the saturated vapor curves, drawn on common coordinates, for several typical refrigerants. If the degree of departure from the vertical is assumed to

be the only factor involved, among the commonly used refrigerants, ammonia would have the largest superheat horn and Freon-12 the smallest. The saturation curve of ethyl ether, a little-used refrigerant, most nearly eliminates the superheat horn.

Theoretically, the most efficient cycle using any refrigerant would be one completely eliminating all superheat by operating with wet compression and thereby discharging dry saturated gas at the end of compression. Practically, however, this is not the case. The unevaporated liquid refrigerant entering the compressor exists in droplet form and not in molecular suspension, and much of it may still remain in droplet form at the end of compression. Superheated gases will be discharged from the compressor, and the liquid droplets will tend to remain in the clearance space. Upon reexpansion of the clearance gases much of this liquid refrigerant will flash into vapor and thus materially increase the volume of the clearance gases. Therefore the volumetric efficiency is greatly reduced, and the theoretical advantages disappear.

Mathematical Analyses of Cycles. In most mathematical analyses of refrigeration cycles many of the differences between theory and practice are disregarded as negligible. Thus any pressure differences across the compressor valves are usually neglected because their effect on calculations is small except at very low evaporator temperatures and pressures. Compression is usually assumed to be isentropic or at least polytropic. The consideration of vapor superheating and subcooling introduces no particular difficulties and therefore is often included. The effects of heat exchanges between parts of the system and the ambient air are difficult to analyze and frequently are small, so that they are usually disregarded. The effect of volumetric efficiency is extremely important and is discussed separately in the following section.

4.7. Volumetric Efficiency. A theoretically perfect compressor would have neither clearance nor losses of any type and would pump on each stroke a quantity of refrigerant equal to the piston displacement. No actual compressor is able to do this, since it is impossible to construct a compressor without clearance or one that will have no wire-drawing through the suction and discharge valves, no superheating of the suction gases upon contact with the cylinder walls, or no leakage of gas past the piston or the valves. All these factors affect the volume of gas pumped or the capacity of the compressor. Some of them affect the horsepower requirements per ton of refrigeration developed.

The ratio of the actual volume of gas drawn into the compressor (at evaporator temperature and pressure) on each stroke to the piston displacement is termed *volumetric efficiency*. If the effect of clearance alone is considered, the resulting expression may be termed *clearance volumetric efficiency*. The expression used for grouping into one constant

all the factors affecting efficiency may be termed *total volumetric efficiency*.² The clearance volumetric efficiency may be calculated with reasonable accuracy; the total volumetric efficiency is best obtained by actual laboratory tests, although a fair approximation to it may be calculated if sufficient data are available.

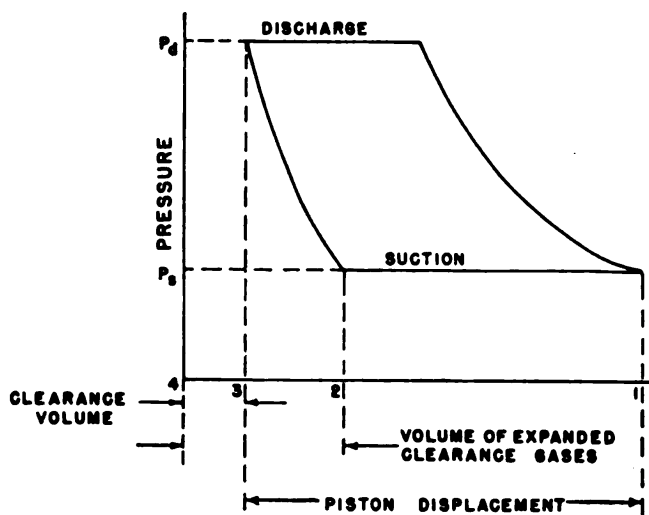


Fig. 4.8.

Clearance Volumetric Efficiency. The volume of space between the end of the cylinder and the piston when the latter is in dead center position is termed *clearance volume*; in Fig. 4.8 this is represented by volume 3-4. The *clearance*, C , is the ratio, expressed in per cent, of the clearance volume to the piston displacement. The piston displacement is shown in Fig. 4.8 as 3-1. Upon expansion of the clearance gases from the discharge pressure, p_d , to the suction pressure, p_s , the volume of these gases will change from 4-3 to 4-2, and the portion of the piston displacement effective in drawing in a new charge of gas from the evaporator will be reduced by 3-2. The ratio of this actual volume of new suction gases to the piston displacement is defined as the *clearance volumetric efficiency*, η_{cv} . Thus,

$$\eta_{cv} = \frac{v_1 - v_2}{v_1 - v_3}$$

² To date various authorities are not in agreement on the best method of subdividing the volumetric efficiency. The expressions here noted are only two of those encountered. Others are *volumetric efficiency due to superheating*, usually found as an empirical equation attempting to evaluate the effect of suction-gas heat absorption from the cylinder walls; *conventional volumetric efficiency* or *theoretical volumetric efficiency*, other terms for the clearance volumetric efficiency; *compression efficiency*, a measure of the deviation of the actual compression process from the adiabatic; *volumetric efficiency without clearance*, an empirical grouping and evaluation of all efficiency factors other than the clearance volumetric efficiency; and the *leakage volumetric efficiency*, an empirical evaluation of the valve and piston-ring leakage.

but

$$v_1 - v_2 = (v_1 - v_3) - (v_2 - v_3)$$

and, since expansion from p_d to p_s is polytropic,

$$v_2 = v_3 \left(\frac{p_d}{p_s} \right)^{\frac{1}{n}}$$

By definition,

$$C = \frac{v_3}{v_1 - v_3}$$

Therefore, by substitution,

$$\begin{aligned} \eta_{cv} &= \frac{(v_1 - v_3) - (v_2 - v_3)}{(v_1 - v_3)} \\ &= 1 - \frac{v_3 \left(\frac{p_d}{p_s} \right)^{\frac{1}{n}} - v_3}{v_1 - v_3} \\ &= 1 + \frac{v_3}{v_1 - v_3} \left[1 - \left(\frac{p_d}{p_s} \right)^{\frac{1}{n}} \right] \\ &= 1 + C - C \left(\frac{p_d}{p_s} \right)^{\frac{1}{n}} \end{aligned} \quad (4.5)$$

This clearance volumetric efficiency has a marked effect upon the compressor piston displacement required per ton of refrigeration developed. The effect becomes more marked as the compression ratio or the spread between condenser and evaporator pressures increases. However, it is usually considered that clearance has no effect upon the horsepower requirements per ton of refrigeration, because the alternate compression and expansion of a fixed quantity of vapor theoretically requires zero power. This process is analogous to the alternate compression and expansion of a spring with the work required during compression recovered during the following expansion.

In an actual system it is probable that clearance has some effect upon the power requirements since the compression and expansion of the clearances gases are not likely to follow exactly the same path; that is, the work required during compression is not numerically identical to the work recovered during expansion. The difference, however, is usually small, and reasonably accurate results are obtained if equation 4.5 is used in calculating compressor displacement requirements and is disregarded in calculating compressor power requirements per ton of refrigeration.

Total Volumetric Efficiency. The total volumetric efficiency of a compressor is best obtained by actual laboratory measurements of the amount of refrigerant compressed and delivered to the condenser. It is too difficult to predict the effects of wire-drawing, cylinder-wall heating, and piston leakage to allow any degree of accuracy in most cases.

An *approximation* to the total volumetric efficiency may be calculated

if the pressure drop through the suction valves and the temperature of the gases at the end of the suction stroke are known and if it is assumed that there is no leakage past the pistons during compression. This approximation may be computed with a modification of equation 4.5; that is, multiplying it by the ratio of the evaporator and cylinder suction pressures and by the evaporator and cylinder suction absolute temperatures. Thus

$$\eta_{iv} = \left[1 + C - C \left(\frac{p_d}{p_s} \right)^{\frac{1}{n}} \right] \times \frac{p_c}{p_s} \times \frac{T_s}{T_c}, \quad (4.6)$$

where the subscript c refers to the compressor cylinder and s refers to the evaporator or the suction line just adjacent to the compressor. The temperature rise of the gases passing from the suction line into the cylinders is difficult to evaluate. If tests are to be made to determine this temperature rise, it usually is found more practical to run complete tests and determine the total volumetric efficiency directly. In some cases reasonable assumptions can be based upon data available from previous tests. Determinations of the suction-valve pressure drops may be made from indicator-card data. The assumption that there is no leakage past the pistons is reasonable for well-designed compressors in good repair. Equation 4.6 applies only to calculations for the determination of the piston displacement required per ton of refrigeration and then only *when v_s and p_s are at evaporator and not cylinder conditions and p_d is at condenser pressure*. If pressures and the specific volume are taken at cylinder conditions, equation 4.5 may be used.

Although the clearance volumetric efficiency has little or no effect upon compressor power requirements, at least a few of the factors grouped in the total volumetric efficiency will affect the compression work. A reasonable approximation to the actual power requirements may be made if they are based upon calculations using the actual cylinder pressures and not the evaporator and condenser pressures, and if the polytropic compression exponent is known and used instead of the adiabatic.

In summary, the most accurate efficiency determinations may be made of a compressor by actual tests. A reasonable approximation to the total volumetric efficiency as it affects compressor displacement may be made by using equation 4.6. Approximations of compressor power requirements may be most accurately made by using the polytropic compression exponent and by basing the work calculations upon the actual cylinder pressures.

4.8. Rotary Compressors. Sections 4.5 and 4.6, which discuss departures from the theoretical vapor-compression refrigeration cycle and volumetric efficiency, are applicable to systems using reciprocating compressors for the compression process. The analyses would remain

essentially the same for systems using rotary compressors, but the evaluation of the factors involved is much more difficult. Clearance values are difficult to determine; cylinder-wall heating must be renamed, although essentially the same superheating process of the suction gases occurs; and it is probably incorrect to assume even polytropic compression. Rotary-compressor capacity and power requirements are best determined by actual calorimeter tests.

4.9. Mathematical Analysis of Vapor Compression Refrigeration.

In §3.15 it was shown that for a constant-pressure process, such as occurs in refrigerant evaporation and condensation in the vapor compression cycle,

$$dQ = dh$$

Therefore [see Fig. 4.4(c)] the amount of heat absorbed during evaporation is

$$Q_{\text{evap}} = (h_3 - h_2) \text{ Btu per lb} \quad (4.7)$$

and the amount of heat rejected during condensation is

$$Q_{\text{cond}} = (h_4 - h_1) \text{ Btu per lb} \quad (4.8)$$

It was also shown in §3.15 that for a throttling process no work is done, and, with approximately equal entering and leaving velocities,

$$h_{\text{entering}} = h_{\text{leaving}}$$

and therefore

$$h_1 = h_2 \quad (4.9)$$

The quality of the refrigerant at the end of expansion is

$$x = \frac{h_1 - h_{f2}}{h_{f2}} \quad (3.21)$$

In accordance with the law of energy conservation, the heat rejected to the condenser must equal the heat absorbed in the evaporator plus the heat equivalent of the work of compression, or

$$Q_{\text{cond}} = Q_{\text{evap}} + \frac{W_{k \text{ comp}}}{J}$$

If the expressions given in equations 4.7, 4.8, and 4.9 are substituted in this equation,

$$\frac{W_{k \text{ comp}}}{J} = (h_4 - h_3) \text{ Btu per lb} \quad (4.10)$$

This equation assumes no heat gains or losses to the ambient air and no heat losses during the compression process.

Refrigerating effect is the amount of heat absorbed by the refrigerant in its travel through the evaporator. In Fig. 4.4(c) this effect is repre-

sented by the expression of equation 4.7. In addition to the latent heat of vaporization it may include any heat of superheat absorbed in the evaporator.

Weight of refrigerant circulated per (minute)(ton of refrigeration) may be determined by dividing the amount of heat in Btu absorbed per (minute)(ton of refrigeration) by the refrigerating effect. Thus

$$\text{Weight refrigerant circulated} = \frac{200}{(h_3 - h_2)} \text{ lb per (ton)(min)} \quad (4.11)$$

Theoretical piston displacement per (ton of refrigeration)(minute) may be found by multiplying the weight of refrigerant to be circulated per (ton of refrigeration)(minute) by the specific volume of the refrigerant gas, $(V_g)_3$, at its entrance to the compressor. Thus

Theoretical piston displacement

$$= \frac{200}{(h_3 - h_1)} (V_g)_3 \text{ cu ft per (ton)(min)} \quad (4.12)$$

Theoretical horsepower per ton of refrigeration is the horsepower theoretically required to compress the refrigerant. It does not take into consideration the mechanical efficiency or the volumetric efficiency of the compressor. Compression of the refrigerant is usually considered to be isentropic, since the degree of deviation is usually not great unless the compressor is cooled by external means.

If isentropic compression is assumed, the heat equivalent of the work of compression per ton of refrigeration is found by taking the product of equations 4.11 and 4.10:

$$\text{Heat equivalent of compression work} = \frac{200 (h_4 - h_3)}{(h_3 - h_1)} \text{ Btu per (ton)(min)}$$

and the horsepower per ton of refrigeration is given by the equation

$$\begin{aligned} \text{Horsepower per ton} &= 200 \left[\frac{(h_4 - h_3)}{(h_3 - h_1)} \right] \left[\frac{778}{33,000} \right] \\ &= 4.717 \frac{(h_4 - h_3)}{(h_3 - h_1)} \end{aligned} \quad (4.13)$$

where 778 is the mechanical equivalent of heat and 33,000 is the foot-pounds per minute horsepower. The horsepower requirements may also be determined for polytropic compression by means of the polytropic, steady-flow work equation shown in Table 3.1 and equation 4.11. Thus

Work of compression

$$= \frac{200}{(h_3 - h_1)} \frac{n}{(n - 1)} (p_3 V_3 - p_4 V_4) \text{ (ft)(lb) per (min)(ton)}$$

and

Theoretical power requirements

$$= \frac{200}{(h_3 - h_1)} \frac{n}{(n - 1)} p_3 V_3 \left[\left(\frac{p_4}{p_3} \right)^{\frac{n-1}{n}} - 1 \right] \frac{144}{33,000} \text{ hp per ton}$$

$$= \frac{0.873 n p_3 V_3 \left[\left(\frac{p_4}{p_3} \right)^{\frac{n-1}{n}} - 1 \right]}{(n - 1) (h_3 - h_1)} \text{ hp per ton} \quad (4.14)$$

If compression is isentropic, $n = \gamma = c_p/c_v$. Values for γ for many common refrigerants are given in Table 4.1.

TABLE 4.1
SPECIFIC HEATS OF REFRIGERANT VAPORS*

Refrigerant	Chemical Symbol	c_p Btu per (lb)(F)	$c_p/c_v = \gamma$
Air.....	0.240	1.40 (-22 to +50 F)
Ammonia.....	NH ₃	0.523	1.31 (70 F)
Carbon dioxide.....	CO ₂	0.206	1.29 (50 F)
Ethane.....	C ₂ H ₆	0.413	1.22 (59 F)
Ethyl chloride.....	C ₂ H ₅ Cl	0.27	1.17 (70 F)
Ethylene.....	C ₂ H ₄	0.360	1.26 (60 F)
Dichlorodifluoromethane } (Freon-12)	CCl ₂ F ₂	0.147	1.13 (50 F)
Methane.....	CH ₄	0.528	1.31 (59 F)
Methyl chloride.....	CH ₃ Cl	0.24	1.28 (77 F)
Methylene chloride.....	CH ₂ Cl	0.154	1.20 (70 F)
Monochlorodifluoromethane } (Freon-22)	CHClF ₂	0.157	1.18 (118 F)
Propane.....	C ₃ H ₈	0.473	1.15 (60 F)
Sulfur dioxide.....	SO ₂	0.154	1.26 (61 to 396 F)
Trichloromonofluoromethane } (Freon-11)	CCl ₃ F	0.138	1.13 (118 F)
Water vapor.....	H ₂ O	0.466	1.27 (32 F)

* One atmosphere pressure. Temperatures shown parenthetically.

If the compressor cylinders are jacketed, an appreciable amount of heat may be rejected to the cooling water during compression. If the suction and discharge compression conditions are known, this heat quantity may be determined from the general energy equation as the difference between the heat equivalent of the work of compression and the difference between the initial and final enthalpies. Thus

Heat rejected to compressor cooling water

$$= \left[\frac{n}{(n - 1)} J (p_3 V_3 - p_4 V_4) - (h_4 - h_3) \right] \text{ Btu per lb} \quad (4.15)$$

Coefficient of Performance. The coefficient of performance for a reversed Carnot-cycle refrigerating machine was derived as

$$\text{c.p.} = \frac{T_2}{T_1 - T_2} \quad (4.2)$$

For any refrigerating machine operating on any cycle,

$$\text{c.p.} = \frac{\text{heat absorbed in evaporator}}{\text{heat equivalent of net work supplied}} \quad (4.16)$$

and, in terms of the enthalpies for the vapor compression cycle with isentropic compression, this equation becomes

$$\text{c.p.} = \frac{h_3 - h_1}{h_4 - h_3} \quad (4.17)$$

It should be noted that equation 4.13, the theoretical horsepower per ton of refrigeration, and equation 4.17, the coefficient of performance, are inverse functions with

$$\text{Horsepower per ton} = \frac{4.717}{\text{c.p.}} \quad (4.18)$$

Heat Removed through Condenser. This term includes all heat removed through the condenser, either as latent heat, heat of superheat, or heat of liquid. Theoretically, it is equivalent to the heat absorbed in the evaporator plus the work of compression. From the product of equations 4.11 and 4.8 it may be expressed as

Heat removed through condenser

$$= 200 \left[\frac{(h_4 - h_1)}{(h_3 - h_1)} \right] \text{ Btu per (ton)(min)} \quad (4.19)$$

Volumetric Efficiency. The volumetric efficiencies were discussed in §4.7, where the clearance volumetric efficiency was derived as

$$\eta_{cv} = 1 + C - C \left(\frac{p_d}{p_s} \right)^{\frac{1}{n}} \quad (4.5)$$

and the total volumetric efficiency was derived as

$$\eta_{tv} = \left[1 + C - C \left(\frac{p_d}{p_s} \right)^{\frac{1}{n}} \right] \times \frac{p_c}{p_s} \times \frac{T_s}{T_c} \quad (4.6)$$

where C is the percentage clearance and the subscripts s and c refer to conditions in the evaporator and the cylinder at the end of the suction stroke respectively.

Actual piston displacement per ton of refrigeration per minute may

be determined by dividing the theoretical piston displacement, equation 4.12, by the total volumetric efficiency, equation 4.6. Thus

$$\text{Actual piston displacement} = \frac{\text{theoretical piston displacement}}{\eta_{iv}} \quad (4.20)$$

If insufficient data are available for evaluation of the total volumetric efficiency, then the clearance volumetric efficiency may be used. However, the resulting required piston displacement will then err on the low side, and a safety factor of from 5 to 25 per cent, depending upon the refrigerant, the compression ratio, and the compressor design, should be applied. With refrigerants having a small superheat horn, such as Freon-12, and with small compression ratios, this factor would tend toward the lower figure; with refrigerants having a large superheat horn, such as ammonia, and with a higher work of compression, this factor will approach the higher figure.

Actual horsepower per ton of refrigeration may be approximated by using the correct value of n in equation 4.14, by using the actual cylinder suction and discharge pressures, and by dividing the resulting horsepower requirements by the mechanical efficiency. The mechanical efficiency is the ratio of the indicated horsepower of the compressor cylinder to the horsepower required to drive the compressor. All three of these factors cannot be calculated. They may be approximated but are best determined by actual test. Indicator cards may be used to determine the indicated horsepower on large compressors; on small compressors the volume of the indicator and its connecting piping is sufficiently large to add materially to the compressor clearance volume and thus reduce the volume of gas pumped and the work required. The actual capacity and power requirements of small condensing units are usually determined by their manufacturers through the aid of calorimeter tests.

EXAMPLE 4.2. A food-storage locker requires a refrigeration system of 12-tons capacity at an evaporator temperature of 20 F and a condenser temperature of 86 F. The refrigerant, ammonia, is subcooled 9 F before entering the expansion valve, and the vapor is superheated 9 F before leaving the evaporator coil. Compression of the refrigerant is adiabatic, and compressor valve throttling and clearance are to be disregarded. A two-cylinder vertical single-acting compressor with stroke equal to 1.5 times the bore is to be used operating at 900 rpm. Determine (a) refrigerating effect, (b) weight of refrigerant to be circulated per minute, (c) theoretical piston displacement per minute, (d) theoretical horsepower, (e) coefficient of performance, (f) heat removed through condenser, and (g) theoretical bore and stroke of compressor.

SOLUTION:

(a) Refrigerating effect = $623.3 - 128.5 = 494.8$ Btu per lb

(b) By equation 4.11,

$$\begin{aligned} \text{Weight of refrigerant circulated} &= \frac{200}{494.8} \\ &= 0.404 \text{ lb per (min)(ton)} \\ (0.404)(12) &= 4.86 \text{ lb per min} \end{aligned}$$

(c) By equation 4.12,

$$\begin{aligned}\text{Theoretical piston displacement} &= (0.404)(6.10) \\ &= 2.46 \text{ cu ft per (min)(ton)} \\ (2.46)(12) &= 29.6 \text{ cu ft per min}\end{aligned}$$

(d) By equation 4.13,

$$\begin{aligned}\text{Theoretical horsepower} &= 4.717 \left(\frac{701.9 - 623.3}{623.3 - 128.5} \right) \\ &= 0.75 \text{ hp per ton} \\ (0.75)(12) &= 9 \text{ hp}\end{aligned}$$

Alternate solution by equation 4.14:

$$\begin{aligned}\text{Theoretical horsepower} &= \frac{(0.873)(1.31)(48.21)(6.1) \left[\left(\frac{169.2}{48.21} \right)^{\frac{0.31}{1.31}} - 1 \right]}{(1.31 - 1)(623.3 - 128.5)} \\ &= 0.75 \text{ hp per ton}\end{aligned}$$

(e) By equation 4.17,

$$\text{c.p.} = \frac{623.3 - 128.5}{701.9 - 623.3} = 6.28$$

(f) By equation 4.19,

$$\begin{aligned}\text{Heat removed through condenser} &= 200 \left(\frac{701.9 - 128.5}{623.3 - 128.5} \right) \\ &= 232 \text{ Btu per (min)(ton)} \\ (232)(12) &= 2784 \text{ Btu per min}\end{aligned}$$

(g) Theoretical cylinder dimensions:

$$\text{Theoretical piston displacement per cylinder} = \frac{29.6}{2} = 14.8 \text{ cu ft per min}$$

$$\text{Pumping capacity per compressor cylinder} = (\text{piston area})(\text{stroke})(\text{rpm})$$

$$\frac{(\pi)(d^3)(1.5)(900)}{(4)(1728)} = 14.8$$

$$\text{Bore} = d = 2.89 \text{ in.}$$

$$\text{Stroke} = 1.5d = 4.34 \text{ in.}$$

EXAMPLE 4.3. If the compressor in Example 4.2 has 2 per cent clearance, determine (a) the clearance volumetric efficiency, (b) corrected piston displacement, and (c) bore and stroke of compressor.

SOLUTION:

(a) By equation 4.5,

$$\begin{aligned}\eta_{cv} &= 1 + 0.02 - 0.02 \left(\frac{169.2}{48.21} \right)^{\frac{1}{1.31}} \\ &= 0.968\end{aligned}$$

(b) By equation 4.20,

$$\text{Piston displacement} = \frac{29.6}{0.968} = 30.6 \text{ cu ft per min}$$

(c) Cylinder dimensions:

$$\text{Piston displacement per cylinder} = \frac{30.6}{2} = 15.3 \text{ cu ft per min}$$

$$\frac{(\pi)(d^3)(1.5)(900)}{(4)(1728)} = 15.3$$

$$\text{Bore} = d = 2.92 \text{ in.}$$

$$\text{Stroke} = 1.5d = 4.38 \text{ in.}$$

EXAMPLE 4.4. If the compressor in Example 4.3 has a suction-valve pressure drop of 4 psi and a discharge-valve pressure drop of 2 psi, determine (a) the piston displacement, (b) horsepower, (c) coefficient of performance, (d) heat removed through condenser, and (e) bore and stroke of compressor.

SOLUTION:

(a) By equation 4.6,

$$\eta_{ts} = 0.968 \left(\frac{44.21}{48.21} \right) = 0.888$$

By equation 4.20,

$$\text{Piston displacement} = \frac{29.6}{0.888} = 33.3 \text{ cu ft per min}$$

ALTERNATE SOLUTION:

By equation 4.5, using cylinder pressures,

$$\begin{aligned} \eta_{cs} &= .1 + 0.02 - 0.02 \left(\frac{171.2}{44.21} \right)^{\frac{1}{1.31}} \\ &= 0.964 \end{aligned}$$

By equation 4.12 and 4.20, using V_s at cylinder suction pressure,

$$\begin{aligned} \text{Piston displacement} &= \frac{(200)(6.65)(12)}{(494.8)(0.964)} \\ &= 33.5 \text{ cu ft per min} \end{aligned}$$

(b) By equation 4.13,

$$\begin{aligned} \text{Horsepower} &= 4.717 \left(\frac{709 - 623.3}{623.3 - 128.5} \right) = 0.82 \text{ hp per ton} \\ (0.82)(12) &= 9.84 \text{ hp} \end{aligned}$$

(c) By equation 4.18,

$$\text{c.p.} = \frac{4.717}{0.82} = 5.76$$

(d) By equation 4.19,

$$\begin{aligned} \text{Heat removed through condenser} &= 200 \left(\frac{709 - 128.5}{623.3 - 128.5} \right) \\ &= 235 \text{ Btu per (ton)(min)} \\ (235)(12) &= 2820 \text{ Btu per min} \end{aligned}$$

(e) Bore and stroke of compressor:

$$\frac{(\pi)(d^3)(1.5)(900)}{(4)(1728)} = \frac{33.5}{2}$$

$$\text{Bore} = d = 3.01 \text{ in.}$$

$$\text{Stroke} = 1.5d = 4.51 \text{ in.}$$

EXAMPLE 4.5. If the compressor of Example 4.4 is water-jacketed so that compression follows the polytropic path $n = 1.2$ and the mechanical efficiency is 80 per cent, determine (a) the theoretical horsepower, (b) actual horsepower, (c) heat rejected to compressor cooling water, and (d) heat rejected to condenser cooling water.

SOLUTION:

(a) By equation 4.14,

$$\begin{aligned}\text{Horsepower} &= \frac{(0.873)(1.2)(44.21)(6.65) \left[\left(\frac{171.2}{44.21} \right)^{\frac{1.2-1}{1.2}} - 1 \right]}{(1.2 - 1)(623.3 - 128.5)} \\ &= 0.79 \text{ hp per ton} \\ (0.79)(12) &= 9.48 \text{ hp}\end{aligned}$$

(b) Actual horsepower:

$$\begin{aligned}\frac{9.48}{0.80} &= 11.86 \text{ hp} \\ \frac{11.86}{12} &= 0.99 \text{ hp per ton}\end{aligned}$$

(c) By equation 4.15,

Heat rejected to compressor cooling water

$$\begin{aligned}&= \frac{n}{(n-1)J} p_3 V_3 \left[\left(\frac{p_4}{p_3} \right)^{\frac{n-1}{n}} - 1 \right] - (h_4 - h_3) \\ &= \frac{(1.2)(144)(44.21)(6.65) \left[\left(\frac{171.2}{44.21} \right)^{\frac{1.2-1}{1.2}} - 1 \right]}{(0.2)(778)} - (676 - 623.6) \\ &= 30.3 \text{ Btu per lb} \\ (30.3)(4.86) &= 147.3 \text{ Btu per min}\end{aligned}$$

(d) By equation 4.19,

Heat rejected through condenser

$$\begin{aligned}&= 200 \left(\frac{676 - 128.5}{623.3 - 128.5} \right) = 221.2 \text{ Btu per (ton) (min)} \\ (221.2)(12) &= 2654 \text{ Btu per min}\end{aligned}$$

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PROBLEMS

4.1. Determine the coefficient of performance for a Carnot-cycle refrigerating machine and for a Carnot-cycle heat pump operating between a condenser temperature of 86 F and an evaporator temperature of (a) 50 F, (b) 30 F, (c) 10 F, and (d) -10 F.

4.2. What is the clearance volumetric efficiency for an ammonia compressor designed with 4 per cent clearance and operating between condenser and evaporator temperatures of (a) 86 F and 40 F, (b) 86 F and 5 F and (c) 86 F and -40 F?

4.3. (a) Solve Problem 4.2 if the clearance is 2 per cent; (b) if the refrigerant is dichlorodifluoromethane (Freon-12) and the clearance is 4 per cent.

4.4. A compressor is designed with 2 per cent clearance and is operated between a condenser temperature of 80 F and an evaporator temperature of 10 F. If the cylinder suction pressure is 3 psi below the evaporator pressure, and the suction gases are superheated 15 F after entering the compressor and before the start of compression, determine the total volumetric efficiency when the refrigerant is (a) ammonia, (b) Freon-12, (c) Freon-22, and (d) methyl chloride.

4.5. A methyl chloride refrigeration system operates between 90 F condensing and 0 F evaporating. If the theoretical cycle of operation with dry compression is assumed, determine (a) refrigerating effect, (b) weight of refrigerant circulated per (min)(ton of refrigeration), (c) piston displacement per (min)(ton of refrigeration), (d) horsepower required per ton of refrigeration, (e) coefficient of performance, and (f) heat removed through condenser per (min)(ton of refrigeration).

4.6. Solve Problem 4.5 if wet compression is assumed.

4.7. Determine the quality of the refrigerant leaving the expansion valve and leaving the evaporator for Problems 4.5 and 4.6.

4.8. A four-cylinder single-acting compressor with 3-in. bore and 4-in. stroke operates at 600 rpm. Freon-12 is the refrigerant, and the condenser and evaporator pressures are 107.9 and 26.51 psia, respectively. If compression is isentropic, the clearance is 2 per cent and there is no subcooling or superheating of the refrigerant, determine (a) refrigerating capacity in tons and also in Btu per hr, (b) theoretical horsepower when operating under full capacity, (c) horsepower when operating under full capacity with $n = 1.17$, (d) heat added during compression when $n = 1.17$, and (e) heat rejected to condenser cooling water per hour.

4.9. Derive equation 4.15 from the general energy equation.

4.10. Saturated ammonia vapor is compressed from an evaporator pressure of 34.27 psia to a condenser pressure of 169.2 psia. The pressure drop is 4 psi through the suction valves and 6 psi through the discharge valves; n is equal to 1.15. Find (a) the error introduced in calculating the work of compression if the valve losses are neglected and if compression is assumed to be isentropic, and (b) heat rejected to the cooling water per pound of refrigerant during the actual compression process.

4.11. What would be the necessary bore and stroke of a single-acting four-cylinder 350-rpm compressor operating upon a simple saturation cycle between the temperatures of 100 and 10 F and developing 12 tons of refrigeration when the

refrigerant is (a) ammonia, (b) methyl chloride, and (c) Freon-11? Assume stroke equal to 1.5 times bore and a total volumetric efficiency of 100 per cent.

4.12. A refrigeration system is operating with 75 F condenser temperature, 10 F evaporator temperature, 5 F subcooling of the liquid refrigerant leaving the condenser, and 7 F superheating of the vapor leaving the evaporator. Compression is isentropic and the clearance is 2 per cent. Neglect valve losses and additional superheating of gases entering the compressor. Determine the required cylinder displacement per (ton of refrigeration)(min) when (a) the refrigerant is Freon-12, (b) the refrigerant is Freon-22, (c) the evaporator temperature is lowered to -40 F and the refrigerant is Freon-12, and (d) the evaporator temperature is 40 F and the refrigerant is Freon-22.

4.13. A Freon-12 refrigerating system is operating with a condensing temperature of 86 F and an evaporating temperature of 25 F. (a) If the liquid line from the condenser is soldered to the suction line from the evaporator to form a simple heat exchanger, and if as a result of this the saturated liquid leaving the condenser is subcooled 6 F, how many degrees will the saturated vapor leaving the evaporator be superheated? Assume no external gain or loss of heat from this heat exchanger. (b) Determine the theoretical coefficient of performance with and without this heat exchanger. Assume isentropic compression.

4.14. A food-freezing system requires 20 tons of refrigeration at an evaporator temperature of -30 F and a condenser temperature of 72 F. The refrigerant, Freon-22, is subcooled 6 F before entering the expansion valve, and the vapor is superheated 7 F before leaving the evaporator. Compression is isentropic, and valve throttling and clearance are to be disregarded. A six-cylinder single-acting compressor with stroke equal to bore is to be used, operating at 1500 rpm. Determine (a) the refrigerating effect, (b) weight of refrigerant to be circulated per minute, (c) theoretical piston displacement per minute, (d) theoretical horsepower, (e) coefficient of performance, (f) heat removed through condenser, and (g) theoretical bore and stroke of compressor.

4.15. The compressor of Problem 4.14 has 3 per cent clearance; suction- and discharge-valve pressure drops of 5 and 7 psia, respectively; polytropic compression with $n = 1.15$; superheating of the vapor after passing the suction valves and before compression of 18 F; and a mechanical efficiency of 75 per cent. Determine (a) clearance volumetric efficiency, (b) total volumetric efficiency, (c) piston displacement per minute, (d) theoretical horsepower, (e) actual horsepower, (f) heat rejected during compression, (g) heat rejected to condenser cooling water, and (h) gallons of cooling water required per minute if temperature rise is 10 F.

CHAPTER 5

Refrigerants

5.1. Introduction. Any substance that absorbs heat through expansion or vaporization may be termed a refrigerant. In the broadest sense of the word the term "refrigerant" is also applied to such secondary cooling mediums as brine solutions or cold water. As commonly interpreted, however, refrigerants include only those working mediums which pass through the cycle of evaporation, recovery, compression, condensation, and liquefaction. Thus, circulating cold mediums are not primary refrigerants, nor are cooling mediums such as ice and solid carbon dioxide.

5.2. Common Refrigerants. Desirable refrigerants are those which possess chemical, physical, and thermodynamic properties that permit their efficient application and service in practical designs of refrigerating equipment. In addition, if the volume of the charge is large, there should be no danger to health and property in case of its escape. A great variety of substances, among which are butane, carbon tetrachloride, ethane, hexane, methane, pentane, propane, and chloroform, have been applied to refrigerating systems but found to be of little practical use. These and similar materials are either highly explosive and flammable or possess other combinations of undesirable properties.

Many of the refrigerants used in the United States during the past few years are listed in Table 5.1, along with their chemical formulas and a few of their more important thermal and physical properties. Those with widest application at present are ammonia, the Freon group, methyl chloride, sulfur dioxide, and water vapor. The fields of application for which the refrigerants listed in Table 5.1 commonly have been selected are summarized in the following sections.

5.3. Air. Air was one of the earliest refrigerants and was widely used even as late as World War I wherever a completely nontoxic medium was needed. Although air is free of cost and completely safe, its low coefficient of performance makes it unable to compete with the present-day nontoxic refrigerants. Only where operating efficiency is secondary, as in aircraft refrigeration (see Chapter 7), does air find any modern application as a refrigerant.

5.4. Ammonia. Ammonia is one of the oldest and most widely used of all refrigerants. It is highly toxic and flammable. It has a boiling point of -28.0°F and a liquid specific gravity of 0.684 at atmospheric pressure. Its greatest application has been in large industrial and commercial reciprocating compression systems where high toxicity is second-

TABLE 5.1
PHYSICAL AND THERMAL PROPERTIES OF COMMON REFRIGERANTS

Refrigerant	Chemical Formula	Boiling Point, F	Freezing Point, F	Critical Point		Specific Gravity of Liquid at Atmos. Press.	Specific Heat of Liquid, Avg. 5 F to 86 F
				Temperature, F	Pressure, lb per sq in. abs.		
Ammonia.....	NH ₃	- 28.0	-107.9	271.2	1651.0	0.684	1.12
Carbon dioxide.....	CO ₂	-109.3	- 69.9	87.8	1069.9	1.560	0.77
Dichlorodifluoromethane (Freon-12).....	CCl ₂ F ₂	- 21.6	-252.0	232.7	582.0	1.480	0.23
Dichloroethylene (Dielen).....	C ₂ H ₂ Cl ₂	118.0	- 70.0	470.0	795.0	1.270	0.27
Dichloromonofluoromethane (Freon-21).....	CHCl ₂ F	48.0	-211.0	353.3	750.0	1.350	0.24
Ethyl chloride.....	C ₂ H ₅ Cl	54.5	-217.7	369.0	764.0	0.43
Isobutane.....	C ₄ H ₁₀	10.0	-229.0	272.7	557.1	0.549	0.62
Methyl chloride.....	CH ₃ Cl	- 10.6	-144.0	289.6	969.2	1.002	0.38
Methyl formate.....	C ₂ H ₄ O ₂	89.2	-147.5	418.0	607.0	0.982	0.51
Methylene chloride (Carrene-1).....	CH ₂ Cl ₂	103.6	-142.0	421.0	640.0	1.291	0.33
Monochlorodifluoromethane (Freon-22).....	CHClF ₂	- 41.4	-256.0	204.8	716.0	1.411	0.30
Sulfur dioxide.....	SO ₂	13.8	- 98.9	314.8	1141.5	1.357	0.34
Trichloromonofluoromethane (Freon-11, Carrene-2).....	CCl ₃ F	74.7	-168.0	388.4	635.0	1.468	0.21
Trichlorotrifluoroethane (Freon-113).....	CCl ₂ F-CClF ₂	117.6	- 31.0	417.4	495.0	1.559	0.21
Water.....	H ₂ O	212.0	32.0	706.1	3226.0	1.000	1.00

ary. For industrial application it has been found valuable because of its low volumetric displacement, low cost, low weight of liquid circulated per ton of refrigeration, and high efficiency. However, in recent years there has been some decrease in the volume of new installations using ammonia. It is also widely used as the refrigerant in absorption systems.

5.5. Carbon Dioxide. Carbon dioxide, a colorless and odorless gas, is heavier than air. It has a boiling point of -109.3°F and a liquid specific gravity of 1.56 at atmospheric pressure. It is nontoxic and nonflammable but has extremely high operating pressures. Because of the high horsepower requirements per ton of refrigeration and the high operating pressures, it has received only limited usage. In former years it was selected for marine refrigeration, for theater air-conditioning systems, and for hotel and institutional refrigeration instead of ammonia because it is nontoxic. The Freon group have largely supplanted it for these applications. At the present time its use is limited primarily to the manufacture of dry ice (solid carbon dioxide). The cycle efficiency can be improved by the use of two-stage compression, and when applied in this manner carbonic refrigeration has received some usage. It has also been used for very low-temperature work by adapting it to a binary or "cascade" cycle, in which the efficiency is improved by using carbon dioxide in the low-temperature stage and ammonia or some other refrigerant in the high-temperature stage.

5.6. The Freons. Freon is the trade name adopted for the following group of refrigerants: trichloromonofluoromethane (Freon-11 or Carrene-2), CCl_3F ; dichlorodifluoromethane (Freon-12), CCl_2F_2 ; monochlorotrifluoromethane (Freon-13), CClF_3 ; tetrafluoromethane (Freon-14), CF_4 ; dichloromonofluoromethane (Freon-21), CHCl_2F ; monochlorodifluoromethane (Freon-22), CHClF_2 ; trichlorotrifluoroethane (Freon-113), $\text{CCl}_2\text{F}-\text{CClF}_2$; and dichlorotetrafluoroethane (Freon-114), $\text{C}_2\text{Cl}_2\text{F}_4$. They are almost universally referred to by their trade names. Freon-11 or Carrene-2 has a boiling point of 74.7°F ; Freon-12, -21.6°F ; Freon-13, -114.5°F ; Freon-14, -198.2°F ; Freon-21, 48.0°F ; Freon-22, -41.4°F ; Freon-113, 117.6°F ; and Freon-114, 38.4°F . The entire group is clear and water-white in color and has a somewhat ethereal odor similar to that of carbon tetrachloride. They are all nonflammable and for all practical purposes nontoxic. Freon-11, or Carrene-2, is widely used for centrifugal refrigeration. In this field it has almost completely supplanted dichloroethylene and methylene chloride (Carrene-1). Freon-12, the most widely used of the Freon group, is generally applied to reciprocating compressors. It has received its widest use in air-conditioning applications where nontoxic, nonflammable refrigerants are required. Freon-21 and Freon-114 have been applied by several manufacturers to rotary compressors in domestic refrigerators. They have also been used experimentally for absorption refrigeration. Freon-22 has been developed for

reciprocating-compressor applications below -20°F , and Freon-13 and Freon-14 are intended for extremely low-temperature usage. Freon-22, however, may receive wide application in higher temperature installations in the future if increased volume of production will permit a lowering of cost. Its favorable characteristics may allow it to compete successfully

TABLE 5.2
COMPOSITION AND BOILING TEMPERATURES* OF HALIDE REFRIGERANTS

Number of Fluorine Atoms	6	Freon-116 C ₂ F ₆ (-108 F)					
	5		Freon-115 C ₂ ClF ₅ (-36 F)				
	4	Freon-14 CF ₄ (-198 F)		Freon-114 C ₂ Cl ₂ F ₄ (38 F)			
	3	Freon-23 (Fluoro- form) CHF ₃ (-116 F)	Freon-13 CClF ₃ (-112 F)		Freon-113 C ₂ Cl ₃ F ₃ (117 F)		
	2	Freon-32 CH ₂ F ₂ (-62 F)	Freon-22 CHClF ₂ (-41 F)	Freon-12 CCl ₂ F ₂ (-22 F)		Freon-112 C ₂ Cl ₄ F ₂ (197 F)	
	1	Freon-41 CH ₃ F (-108 F)	Freon-31 CH ₂ ClF (16 F)	Freon-21 CHCl ₂ F (48 F)	Freon-11 CCl ₃ F (75 F)		Freon-111 C ₂ Cl ₅ F (279 F)
	0	Methane CH ₄ (-250 F)	Methyl chloride CH ₃ Cl (-11 F)	Methyl- ene chloride CH ₂ Cl ₂ (104 F)	Chloro- form CHCl ₃ (142 F)	Carbon tetra- chloride CCl ₄ (171 F)	Hexa- chloro- ethane C ₂ Cl ₆ (365 F)
		0	1	2	3	4	5

Number of Chlorine Atoms

* Boiling temperatures at atmospheric pressure shown parenthetically.

with ammonia and possibly with Freon-12 for general refrigeration purposes.

Table 5.2 shows the relationship of halogen content and boiling temperatures for most of the commonly used halide refrigerants. As was pointed out in §5.2, some of the compounds, such as carbon tetrachloride, chloroform, and methane, have been found unsatisfactory for general use as refrigerants although they have been used for this purpose experimentally. Others, such as methyl chloride and some of the Freon group, are widely applied as refrigerants. The Freon group have all been synthesized expressly for refrigeration purposes. It may be concluded from an analysis of this table that there are still many untried possibilities for the development of synthesized refrigerants in either the halide or other groups.

5.7. Methyl Chloride. Methyl chloride, CH_3Cl , is a colorless liquid with a faint, sweet, nonirritating odor. Methyl chloride was introduced about 1920 in the United States for refrigeration purposes and is now widely used. It replaced ammonia and carbon dioxide for many new installations in the 1920's and early 1930's and was widely used during World War II as a substitute for Freon, then unavailable. Its use, however, appears to be definitely on the decline. It has a boiling point of -10.6°F and a liquid specific gravity of 1.002 at atmospheric pressure. It is to a certain degree both flammable and toxic. Methyl chloride has been used in domestic units with both reciprocating and rotary compressors and in commercial units with reciprocating compressors up to approximately 10-tons capacity.

5.8. Sulfur Dioxide. Sulfur dioxide, SO_2 , a colorless gas or liquid, is extremely toxic and has a pungent irritating odor. It is nonexplosive and nonflammable, and has a boiling point of 13.8°F and a liquid specific gravity of 1.36. It is at present one of the most prevalent of all refrigerants in domestic systems, primarily because of the large number of sulfur dioxide domestic units built in the 1930's. It has been applied to both reciprocating and rotary compressors. With such applications the volume of refrigerant charge is small, and there is little danger of fatal concentrations resulting through refrigerant leakage. Sulfur dioxide has also been used to a considerable extent in small-tonnage commercial machines. However, the volume of new units using sulfur dioxide as a refrigerant is small.

5.9. Water Vapor. Water vapor, H_2O , is the cheapest and probably the safest of all refrigerants. However, because of its high freezing temperature of 32°F , it is limited in application to high-temperature refrigeration. Its application has been to steam-jet refrigeration and to centrifugal compression refrigeration. It is, of course, nontoxic, nonflammable, and nonexplosive. Because of its high-temperature limitations and its complete safety, it has been used principally for comfort

air-conditioning applications and to some extent for water cooling. The physical and thermal properties of water vapor are well known and hence are not discussed further in this chapter.

5.10. Hydrocarbon Refrigerants. Many of the hydrocarbons are used as refrigerants in industrial installations, where they are frequently available at low cost. These include butane (C_4H_{10}), isobutane (C_4H_{10}), propane (C_3H_8), propylene (C_3H_6), ethane (C_2H_6), and ethylene (C_2H_4). However, they are all highly flammable and explosive, and therefore their use has been limited principally to the chemical and refining industries, where similar hazards already exist. They all possess satisfactory thermodynamic properties.

Several of the hydrocarbons, including propane, ethane, and ethylene, exhibit promise for use as refrigerants at -100 F or lower. Ethylene has a saturation temperature of -176.8 F at a pressure of 6.75 psia and makes an excellent refrigerant when used in a cascade system with an auxiliary cycle.

Isobutane was used by one manufacturer of domestic refrigerators in a rotary compressor until 1933 but has no such application at the present time. Propane has received limited use as both the motive fuel and the refrigerant in a refrigerated truck unit. Such a cycle is termed transitory, the propane first passing through the evaporator and then to the engine.

5.11. Halogenated Hydrocarbons and Other Refrigerants. Chemical compounds formed from methane (CH_4) and ethane (C_2H_6) by the substitution of chlorine, fluorine, or both, for part of their hydrogen content are termed halogenated hydrocarbons. Many refrigerants, including methyl chloride and the Freons, are included in this group. Some of the others are described below.

Dichloroethylene (Dielene), $C_2H_2Cl_2$, is a colorless liquid with a boiling point of 118.0 F and a liquid specific gravity of 1.27. It has a strong, nonirritating odor similar to that of chloroform and is to a limited extent both toxic and explosive. Its principal application has been in centrifugal compression systems, in which it received limited usage in the 1920's. In recent years it has been supplanted by methylene chloride and trichloromonofluoromethane.

Ethyl chloride, C_2H_5Cl , is a colorless liquid with a boiling point of 54.5 F. It is to a certain degree both toxic and flammable, and is similar in many respects to methyl chloride but with lower operating pressures. It is not used in refrigerating equipment at the present time but has in the past been selected for both rotary and reciprocating compressors.

Methylene chloride (Carrene-1), CH_2Cl_2 , is a clear, water-white liquid with a sweet, nonirritating odor similar to that of chloroform. It has a boiling point of 103.6 F and a liquid specific gravity of 1.291. It is nonflammable and nonexplosive and is toxic only at comparatively high

concentrations. Methylene chloride is used by several manufacturers in domestic rotary compressors and has found some use commercially for absorption refrigeration because of its low toxicity and absence of fire hazard. It has been used successfully in centrifugal compressors but has been supplanted in recent years by Freon-11 (Carrene-2).

Methyl formate, $C_2H_4O_2$, not one of the halogenated hydrocarbons, is a relatively new refrigerant. It has a boiling point of 89.2 F and a liquid specific gravity of 0.982 at atmospheric pressure. It is highly toxic, flammable, and explosive. It has been used to some extent in domestic-refrigeration rotary compressors.

5.12. Comparison of Refrigerants. In the choice of a refrigerant it should be remembered that as yet no one substance has proved the ideal working medium under all operating conditions. The characteristics of some refrigerants made them desirable for use with reciprocating compressors. Other refrigerants are best adapted to centrifugal or rotary compressors. In some applications toxicity is of negligible importance, whereas in others, such as comfort cooling, a nontoxic and nonflammable refrigerant is essential. The requirements of a refrigerant to be used for low-temperature work are different from those for high-temperature applications. Therefore in selecting the correct refrigerant it is necessary to determine those properties which are most desirable and to choose the one most closely approaching the ideal for the particular application. This section compares the respective merits of the common refrigerants.

Evaporator and Condenser Pressures. It is desirable that both condenser and evaporator pressures be positive, yet not too high above atmospheric pressure. Positive pressures prevent leakage of air and moisture into the system and also make it easier to detect leaks. However, if the pressures are high, heavy construction of compressor, evaporator, and condenser is usually necessary, resulting in an increased initial cost and a less compact system. High operating pressures also tend to increase power consumption.

Figure 5.1 shows graphically the pressure-temperature relationships for refrigerants when operating between -60 F and 200 F. Table 5.3 presents evaporator and condenser pressures, operating-pressure differences, and compression ratios for various refrigerants when operating on the standard cycle of 5 F evaporator and 86 F condenser, and also on the cycle between 40 F evaporator and 100 F condenser. The refrigerants in this table are arranged in order of increasing pressures. Carbon dioxide is the only refrigerant having operating pressures sufficiently high to offer any great structural difficulty for equipment.

Almost any refrigerant can be used in conjunction with any type of compressor if proper attention is given to design. However, reciprocating compressors are inherently well adapted for use with refrigerants having low specific volumes, high operating pressures, and high operating-

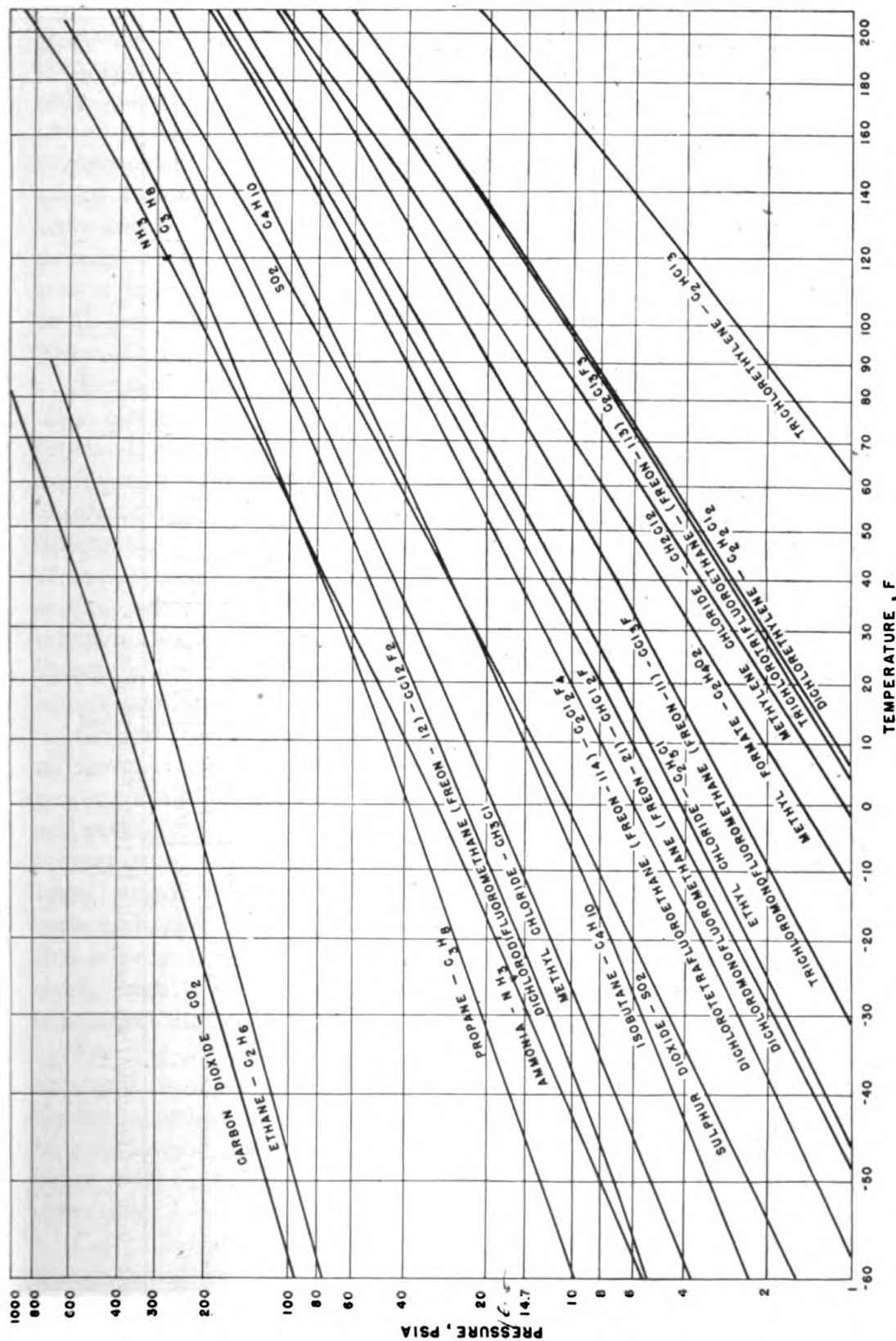


Fig. 5.1. Pressure-temperature relationships of refrigerants. From *Refrigerating Data Book*, 5th ed., 1943, p. 55. American Society Refrigerating Engineers.

TABLE 5.3
CONDENSER AND EVAPORATOR PRESSURES¹

Refrigerant	5 F Evaporator and 86 F Condenser				40 F Evaporator and 100 F Condenser			
	Evaporator Pressure, psia	Condenser Pressure, psia	Pressure Differential, psia	Ratio of Condenser to Evaporator Pressure	Evaporator Pressure, psia	Condenser Pressure, psia	Pressure Differential, psia	Ratio of Condenser to Evaporator Pressure
Trichlorotrifluoroethane (Freon-113).....	0.98	7.86	6.88	8.02	2.66	10.48	7.82	3.93
Methylene chloride (Carrene-1).....	1.17	10.60	9.43	9.07	3.38	13.25	9.87	3.92
Methyl formate.....	1.75	13.81	12.06	7.90	4.66	18.26	13.60	3.92
Trichloromonofluoromethane (Freon-11, Carrene-2).....	2.93	18.28	15.35	6.24	7.03	23.60	16.57	3.36
Ethyl chloride.....	4.65	27.10	22.45	5.83	10.79	34.79	24.00	3.22
Dichloromonofluoromethane (Freon-21).....	5.24	31.2	25.96	5.83	12.32	40.04	27.72	3.25
Isobutane.....	13.1	59.50	46.4	4.54	26.9	73.7	46.8	2.74
Sulfur dioxide.....	11.81	66.45	54.64	5.63	27.10	84.52	57.42	3.12
Methyl chloride.....	21.15	94.70	73.55	4.47	43.25	116.7	73.4	2.70
Dichlorodifluoromethane (Freon-12).....	26.51	107.9	81.39	4.07	51.68	131.6	79.92	2.54
Ammonia.....	34.27	169.2	134.93	4.94	73.32	211.9	138.58	2.89
Monochlorodifluoromethane (Freon-22).....	43.02	174.5	131.5	4.06				
Carbon dioxide.....	331.9	1043.0	711.1	3.16				

¹ Adapted from *Refrigerating Data Book*, 5th ed., 1943, Table 10, p. 65.

pressure differentials. With reciprocating compressors the displacement is positive but is limited by the cylinder size and piston speed. Such compressors are capable of developing high pressures but can handle only relatively low refrigerant volumes. However, high evaporator pressures automatically result in low specific volumes at the beginning of compression. Because of the definite temperature-pressure relationship at saturation, refrigerants associated with high operating pressures also have low boiling points at atmospheric pressure. Thus the refrigerants usually selected for reciprocating compressors are the ones with boiling points below 30° F and with pressure differentials of approximately 50 lb per square inch or higher. Carbon dioxide, ammonia, Freon-12, Freon-22, methyl chloride, and sulfur dioxide come within this group.

Because centrifugal compressors do not offer positive displacement, they are best adapted to refrigerants operating at low pressure differentials and under low evaporator and condenser pressures. They are, however, capable of moving large volumes of refrigerant gas. Refrigerants such as Freon-11 and methylene chloride, with operating-pressure differentials of less than 22 lb per square inch and with low-side pressures of approximately 3 lb per square inch or less, are generally used with centrifugal compression. Rotary compressors are best adapted to refrigerants with intermediate pressures and pressure differentials. Pressure differentials falling between approximately 20 and 30 lb per square inch are commonly used.

Critical Temperature and Pressure. If the critical temperature of a refrigerant is too near the desired condensing temperature, excessive power consumption results. Table 5.4 presents the critical temperatures and pressures for the common refrigerants arranged in order of decreasing temperature. With the exception of carbon dioxide, all critical temperatures shown are well above normal condensing temperatures. For carbon dioxide, however, the critical temperature is 87.8° F; and since this is only slightly higher than most condensing temperatures, excessively high pressures result, which necessitate heavy construction of equipment and cause excessive power consumption.

Freezing Temperatures. Table 5.5 shows the common refrigerants arranged in order of increasing freezing temperature. The refrigerant chosen must have a freezing point well below any temperature that is to be encountered in operation. Since all freezing temperatures shown are below -30° F, this requirement becomes a factor only in low-temperature operation.

Cost. In fractional-tonnage systems requiring only a small refrigerant charge, refrigerant cost becomes a relatively unimportant factor. However, in large systems both the initial and the maintenance costs are influenced by the tightness of construction and the leakage tendency of the refrigerant. The rate of leakage is known to be inversely proportional

TABLE 5.4
REFRIGERANT CRITICAL TEMPERATURES AND PRESSURES

Refrigerant	Critical Point	
	Temperature, F	Pressure, psia
Water.....	706.1	3226.
Dichloroethylene (Dielene).....	470.0	795.
Methylene chloride (Carrene-1).....	421.0	640.0
Methyl formate.....	418.0	607.0
Trichlorotrifluoroethane (Freon-113).....	417.4	495.
Trichloromonofluoromethane (Freon-11, Carrene-2).....	388.4	635.0
Ethyl chloride.....	369.0	764.0
Dichloromonofluoromethane (Freon-21).....	353.3	750.
Sulfur dioxide.....	314.8	1141.5
Methyl chloride.....	289.6	969.2
Isobutane.....	272.7	557.1
Ammonia.....	271.2	1651.
Dichlorodifluoromethane (Freon-12).....	232.7	582.0
Monochlorodifluoromethane (Freon-22).....	204.8	716.0
Carbon dioxide.....	87.8	1069.9

TABLE 5.5
REFRIGERANT FREEZING TEMPERATURES

Refrigerant	Freezing Point, F
Monochlorodifluoromethane (Freon-22).....	-256.0
Dichlorodifluoromethane (Freon-12).....	-252.4
Isobutane.....	-229.0
Ethyl chloride.....	-217.7
Dichloromonofluoromethane (Freon-21).....	-211.0
Trichloromonofluoromethane (Freon-11—Carrene-2).....	-168.0
Methyl formate.....	-147.5
Methyl chloride.....	-143.7
Methylene chloride (Carrene-1).....	-142.0
Ammonia.....	-107.9
Sulfur dioxide.....	-98.9
Dichloroethylene (Dielene).....	-70.0
Carbon dioxide.....	-69.9
Trichlorotrifluoroethane (Freon-113).....	-31.0

to the square root of the molecular weight, besides being dependent upon such factors as pressure difference, diffusion rate, viscosity, density, and capillarity.

Coefficient of Performance and Power Requirements. The Carnot-cycle coefficient of performance is 5.74 for an ideal refrigerant operating between 86 F condenser and 5 F evaporator. The coefficient of performance for the common refrigerants operating between these same temperatures on the vapor refrigeration cycle are arranged in decreasing order in Table 5.6. The last column of this table also shows the theoretical horsepower

required per standard ton. If the volumetric and thermal efficiencies remain constant, the horsepower required per standard ton is inversely proportional to the coefficient of performance.

TABLE 5.6
COEFFICIENT OF PERFORMANCE AND POWER REQUIREMENTS FOR VARIOUS
REFRIGERANTS

Refrigerant	Coefficient of Performance (Standard Cycle 86 F Condenser 5 F Evaporator)	Efficiency (% Carnot Cycle)	HP per ton (Standard Cycle 86 F Condenser 5 F Evaporator)
Carnot cycle.....	5.74	100.0	0.82
Dichloroethylene (Dielene).....	5.14	89.4	0.92
Dichloromono fluoromethane (Freon-21)....	5.09	88.8	0.93
Trichloromono fluoromethane (Freon-11, Carrene-2).....	5.14	87.8	0.94
Methyl chloride.....	4.90	85.3	0.96
Methylene chloride (Carrene-1).....	4.90	85.3	0.96
Ammonia.....	4.85	84.5	0.99
Trichlorotrifluoroethane (Freon-113).....	4.79	83.5	1.00
Sulfur dioxide.....	4.73	82.5	1.00
Dichlorodifluoromethane (Freon-12).....	4.72	82.2	1.00
Monochlorodifluoromethane (Freon-22).....	4.56	79.4	1.04
Carbon dioxide.....	2.56	44.6	1.84

Practically all refrigerants in common use have approximately the same coefficients of performance and horsepower requirements. The one exception shown in Table 5.6 is carbon dioxide, with a power requirement of 1.84 hp per ton and a coefficient of performance of 2.56. These poor properties may be attributed directly to carbon dioxide's very low critical point.

Vapor and Liquid Refrigerant Densities. Low refrigerant densities are generally preferable, since they permit the use of smaller suction and discharge lines without excessive pressure drops. With centrifugal compressors, however, a high vapor density is desirable because the centrifugal force imparted to the gas is directly proportional to the vapor density.

Since the velocities in liquid lines are comparatively low, no great advantage is gained by using a low-density refrigerant to obtain a lower pressure drop. However, if the evaporator is located on a different level than the condensing unit, the static head of the refrigerant in the liquid line will be smaller with a low-density liquid. An appreciable static head will result in a reduction in pressure on the high side of the expansion

valve. This pressure reduction will cause a portion of the liquid to boil and thus will require that the expansion valve handle both liquid and gas. To overcome this difficulty, liquid heat exchangers are frequently installed so that the refrigerant is subcooled before entering the liquid line. Table 5.7 shows vapor and liquid densities for practical refrigerants arranged in order of decreasing vapor density.

TABLE 5.7
REFRIGERANT VAPOR AND LIQUID DENSITIES

Refrigerant	Vapor Density, lb per cu ft		Liquid Density at 86 F, lb per cu ft
	5 F	86 F	
Carbon dioxide.....	3.741	21.09	37.4
Monochlorodifluoromethane (Freon-22)...	0.803	3.212	73.4
Dichlorodifluoromethane (Freon-12).....	0.673	2.569	80.6
Methyl chloride.....	0.221	0.925	55.8
Isobutane.....	0.156	0.658	34.1
Sulfur dioxide.....	0.115	0.844	84.4
Ammonia.....	0.122	0.564	37.2
Dichloromonofluoromethane (Freon-21)...	0.110	0.577	84.5
Trichloromonofluoromethane (Freon-11, Carrene-2).....	0.082	0.446	91.4
Ethyl chloride.....	0.058	0.299	55.0
Trichlorotrifluoroethane (Freon-113).....	0.037	0.257	97.0
Methyl formate.....	0.021	0.141	61.0
Methylene chloride (Carrene-1).....	0.020	0.150	83.3

Latent Heat. A high latent heat at the evaporator temperature is desirable because it is usually associated with a high refrigerating effect per pound of refrigerant circulated. This high refrigerating effect in turn reduces the required weight of refrigerant to be circulated per ton of refrigeration. Table 5.8 gives the refrigerating effect for various refrigerants operating on the standard cycle. In addition, the latent heat, the weight of refrigerant circulated per ton, and the volume of the liquid refrigerant circulated per ton are shown.

It is sometimes found on fractional-tonnage equipment that the circulation of a low weight of refrigerant is detrimental from the standpoint of control operation. Greater refrigerant volumes necessitate less sensitive controls, and the adjustments are generally less critical.

Specific Volume. The vapor displacement required of a compressor is dependent upon the specific volume of the refrigerant gas at evaporator temperature and the refrigerating effect per pound of refrigerant. A low vapor displacement per ton of refrigeration is necessary for satisfactory operation of reciprocating compressors because of the comparatively small displacement available. Centrifugal compressors operate satis-

factorily with refrigerants requiring very high vapor displacements and with rotary compressors with refrigerants requiring intermediate vapor displacements.

TABLE 5.8
REFRIGERATING EFFECT AND QUANTITY OF REFRIGERANT REQUIRED PER TON

Refrigerant	Refrigerating Effect, Btu per lb (Standard Cycle 5 F to 86 F)	Latent Heat of Vaporization at 5 F, Btu per lb	Weight Refrigerant Circulated per Standard Ton, lb per min	Volume Liquid Refrigerant Circulated per Standard Ton, cu in. per min
Ammonia.....	474.5	565.0	0.422	19.6
Methyl formate.....	189.2	231.0	1.056	29.9
Methyl chloride.....	150.3	180.7	1.330	41.2
Sulfur dioxide.....	141.4	169.4	1.414	28.9
Methylene chloride (Carrene-1).....	134.1	162.1	1.492	30.9
Isobutane.....	111.5	159.5	1.794	91.1
Dichloromonofluoromethane (Freon-21).....	89.4	109.3	2.237	45.7
Monochlorodifluoromethane (Freon-22).....	69.5	93.5	2.877	67.4
Trichloromonofluoromethane (Freon-11, Carrene-2).....	67.5	84.0	2.961	56.0
Carbon dioxide.....	55.5	117.6	3.603	166.5
Trichlorotrifluoroethane (Freon-113).....	53.7	70.6	3.726	66.5
Dichlorodifluoromethane (Freon-12).....	51.1	69.5	3.916	83.9

Table 5.9 presents the specific volumes, theoretical piston displacements, and boiling points of the common refrigerants. These are arranged in order of increasing specific volume of the refrigerant vapor at 5 F. The required vapor displacement is almost directly related to the boiling point of the refrigerant at atmospheric pressure. Refrigerants having low boiling points are usually operated with reciprocating compressors; refrigerants having intermediate boiling points, with rotary compressors; and refrigerants having high boiling points, with centrifugal compressors.

Stability and Inertness. An ideal refrigerant should not decompose at any temperature normally encountered and should not form higher boiling-point liquids or solid substances through polymerization. Some refrigerants used in both compression and absorption systems tend to disintegrate, forming gases noncondensable within the operating-pressure range and causing high condensing pressures and vapor lock. Disintegration of refrigerants is brought about through disassociation, reaction with metals, or the presence of unsaturated hydrocarbons in the lubricating oil.

It is desirable that a refrigerant be inert with respect to all materials with which it comes in contact. Care must be taken in the choice of packing and of such parts as gaskets. The winding coverings and electrical insulations of hermetically sealed units must be carefully selected.

TABLE 5.9
SPECIFIC VOLUME AND THEORETICAL PISTON DISPLACEMENT FOR
VARIOUS REFRIGERANTS

Refrigerant	Specific Volume of Vapor at 5 F, cu ft per lb	Boiling Point F (Atmos- pheric Pressure)	Vapor Displace- ment per Standard Ton, cu ft per min	Ratio Vapor Displace- ment to That for Carbon Dioxide
Carbon dioxide.....	0.267	-109.3	0.943	1.0
Monochlorodifluoromethane (Freon-22).....	1.24	-41.4	3.56	3.8
Dichlorodifluoromethane (Freon-12).....	1.49	-21.6	5.82	6.2
Methyl chloride.....	4.52	-10.6	6.09	6.5
Isobutane.....	6.41	10.0	11.5	12.2
Sulfur dioxide.....	6.45	13.8	9.08	9.6
Ammonia.....	8.19	-28.0	3.44	3.7
Dichloromonofluoromethane (Freon-21).....	9.09	48.0	20.4	21.6
Trichloromonofluoromethane (Freon-11, Carrene-2).....	12.2	74.7	36.3	38.5
Trichlorotrifluoroethane (Freon-113).....	27.1	117.6	100.8	106.9
Methyl formate.....	47.6	89.2	49.9	52.9
Methylene chloride (Carrene-1).....	50.0	103.6	74.50	79.0

Under certain conditions ammonia will decompose in either compression or absorption systems. The exact cause of the decomposition is unknown, but it may possibly be attributed to the release of hydrogen through the reaction of moisture and metals.

Freon is stable unless exposed to temperatures over 1000 F. At such high temperatures decomposition occurs, forming both corrosive and poisonous products. Since such high temperatures are not reached under normal circumstances, this behavior is not necessarily detrimental. The Freon refrigerants cannot be used with gasket material containing natural rubber, and they act as a solvent with certain types of insulating materials and varnishes. However, they may be used with either neoprene or chloroprene rubber, with pressed fibers having an insoluble binder, with metallic gaskets, and with many synthetic resins.

Methyl chloride, along with most of the halogenated hydrocarbons (those containing either chlorine or fluorine, or both), is comparatively

stable. However, methyl chloride acts as a solvent on many organic materials and composition gaskets in the same manner as Freon. It cannot be used in the presence of natural rubber or even with some of the neoprene compositions.

Sulfur dioxide does not decompose until a temperature of about 3000 F is reached and therefore is one of the most stable of the refrigerants. At temperatures normally encountered there is no decomposition by contact with metals, but at high temperatures there may be some with lubricating oils.

Corrosive Properties. Extreme care must be taken that the compressor chosen for use with a given refrigerant contains no metal with which that refrigerant reacts. A summary of the conditions favoring corrosion for five of the common refrigerants is presented in Table 5.10.

TABLE 5.10
CORROSIVE PROPERTIES OF REFRIGERANTS

<i>Refrigerant</i>	<i>Conditions Favoring Corrosion</i>
Ammonia.....	Pure anhydrous ammonia will dissolve copper unless protected by oil film. Galvanized or tinned surfaces sometimes affected
Carbon dioxide.....	Not corrosive, except to iron and copper and then only when both moisture and oxygen are present
Freon refrigerants.....	Not corrosive to any metals commonly used
Methyl chloride.....	Corrosive to zinc, aluminum, and magnesium alloys in the presence of only small amounts of water
Sulfur dioxide.....	Corrosive to most metals when water is present. Moisture content in system should not exceed fifty parts per million

The Freon refrigerants can be used with practically all metals without danger of corrosion, and ammonia may be used with iron or steel but not with copper or copper compositions. Methyl chloride should not be used in any system containing aluminum, zinc, or magnesium parts. Sulfur dioxide is noncorrosive to all metals in the absence of water. However, since sulfur dioxide reacts with water to form sulfuric acid, care must be taken that the system contains not more than 50 parts of moisture per million by weight. Although corrosion will probably not result until the concentration is above 300 parts per million, sulfur dioxide evaporates more readily than water and thereby allows concentration of moisture in the refrigerant liquid. Carbon dioxide is corrosive to iron and copper if both moisture and oxygen are present. However, corrosion no longer occurs after the oxygen is consumed.

Dielectric Strength. The electrical resistance of a refrigerant is important if it is to be used in a hermetically sealed unit with the motor exposed

to the refrigerant. The relative dielectric strength of the refrigerant is the ratio of the dielectric strength of the nitrogen-vapor mixture to that of nitrogen at atmospheric pressure and room temperature when measured in the same gap. Table 5.11 presents in decreasing order the relative dielectric strengths of many of the common refrigerants. The Freons exhibit the best qualities from this standpoint.

TABLE 5.11
RELATIVE DIELECTRIC STRENGTH OF REFRIGERANT GASES²

<i>Refrigerant</i>	<i>Relative Dielectric Strength</i>
Trichloromonofluoromethane (Freon-11, Carrene-2).....	3.0
Trichlorotrifluoroethane (Freon-113).....	2.6
Dichlorodifluoromethane (Freon-12).....	2.4
Sulfur dioxide.....	1.9
Dichloromonofluoromethane (Freon-21).....	1.33
Methylene chloride (Carrene-1).....	1.11
Methyl chloride.....	1.06
Ethyl chloride.....	1.00
Carbon dioxide.....	0.88
Ammonia.....	0.82

Viscosity. Table 5.12 shows a summary of the liquid and vapor viscosities for the common refrigerants within the temperature range of 5 to 150 F for the vapors and of 5 to 100 F for the liquids. It is desirable that both the liquid and the vapor refrigerants have low viscosities because of the lower pressure drops in passing through liquid and suction lines. Moreover, the heat transfer through condenser and evaporator surfaces is improved at low viscosities.

Thermal Conductivity. For efficient use of both evaporator and condenser surfaces, a refrigerant should possess a high thermal conductivity. Information concerning the thermal conductivities of refrigerants is incomplete. Table 5.13 presents the available values for the common refrigerants in both the liquid and the vapor phases.

Although thermal conductivity is one of the important refrigerant qualities affecting the efficient use of evaporator and condenser surfaces, there are other qualities equally important. Some of these are the surface wetting characteristics of the refrigerant, the solubility and viscosity of the lubricating oil used in the system, and the liquid and vapor velocities. High velocities tend to break down surface film coefficients and increase the rate of heat transfer. If chlorine is contained in the refrigerant, it serves as a wetting agent that tends to form a better contact between the refrigerant and the surface, thus increasing the heat transfer. The chlorinated hydrocarbon refrigerants and the Freon group come within this classification.

² From Charlton, E. E., and Cooper, F. S., "Dielectric Strength of Insulating Fluids." *General Electric Review*, Vol. 40, No. 9 (September, 1937), p. 438.

TABLE 5.12
LIQUID AND VAPOR REFRIGERANT VISCOSITIES IN CENTIPOISES³

Refrigerant	Vapor Temperatures					Liquid Temperatures				
	5 F	20 F	40 F	100 F	150 F	5 F	20 F	40 F	80 F	100 F
Ammonia.....	0.0085	0.0088	0.0093	0.0105	0.0116	0.250	0.240	0.230	0.210	0.200
Trichlorotrifluoroethane (Freon-113).....	0.0093	0.0095	0.0098	0.0106	0.0112	1.200	1.043	0.876	0.646	0.564
Methyl chloride.....	0.0095	0.0098	0.0102	0.0115	0.0128	0.295	0.290	0.268	0.250	0.230
Trichloromonofluoromethane (Freon-11, Car- rene-2).....	0.0096	0.0099	0.0103	0.0113	0.0121	0.650	0.586	0.517	0.417	0.380
Dichloromonofluoromethane (Freon-21).....	0.0102	0.0105	0.0108	0.0118	0.0125	0.470	0.436	0.397	0.337	0.314
Sulfur dioxide.....	0.0111	0.0115	0.0120	0.0134	0.0154	0.503	0.390	0.350	0.290	0.260
Dichlorodifluoromethane (Freon-12).....	0.0114	0.0116	0.0119	0.0129	0.0136	0.328	0.308	0.286	0.255	0.242
Monochlorodifluoromethane (Freon-22).....	0.0114	0.0118	0.0122	0.0133	0.0143	0.286	0.271	0.256	0.232	0.223
Carbon dioxide.....	0.0132	0.0135	0.0140	0.0155	0.0165	0.115	0.110	0.095	0.064	0.680
Water.....	1.540	0.860

³ Adapted from *Refrigerating Data Book*, 5th ed., 1943, Table 7, p. 59.

Oil Effect. Practically all refrigerants are miscible with mineral lubricating oils to a certain degree, dependent upon the operating temperatures and pressures, the viscosity and base of the oil, and the design of the lubricating system. Refrigerants relatively nonmiscible with mineral oil include carbon dioxide, ammonia, sulfur dioxide, and methyl formate. Highly miscible refrigerants include Freon-11, Freon-12, Freon-21, Freon-22, Freon-113, methylene chloride, isobutane, ethyl chloride, and methyl chloride.

TABLE 5.13
THERMAL CONDUCTIVITY OF REFRIGERANTS

Phase	Refrigerant	Temperature F	Thermal Conductivity, (Btu)(ft) per (sq ft)(hr)(F)
Vapor	Ammonia	32	0.0128
	Carbon dioxide	32	0.0081
	Isobutane	32	0.0080
	Monochlorodifluoromethane (Freon-22)	86	0.0068
	Dichloromonofluoromethane (Freon-21)	86	0.0057
	Dichlorodifluoromethane (Freon-12)	86	0.0056
	Methyl chloride	32	0.0049
	Trichloromonofluoromethane (Freon-11, Carrene-2)	86	0.0048
	Trichlorotrifluoroethane (Freon-113) ($\frac{1}{2}$ atmosphere)	86	0.0045
	Sulfur dioxide	32	0.0045
	Methylene chloride (Carrene-1)	32	0.0036
Liquid	Ammonia	14 to 68	0.290
	Sulfur dioxide	68	0.200
	Carbon dioxide	68	0.120
	Methyl chloride	68	0.093
	Dichloroethylene (Dielene)	68	0.089
	Dichloromonofluoromethane (Freon-21)	104	0.067
	Trichloromonofluoromethane (Freon-11, Carrene-2)	104	0.059
	Monochlorodifluoromethane (Freon-22)	104	0.056
	Trichlorotrifluoroethane (Freon-113)	104	0.050
	Dichlorodifluoromethane (Freon-12)	104	0.047

Miscibility should be considered a secondary rather than a determining factor in the choice of a refrigerant. However, with nonmiscible refrigerants larger heat-transfer surfaces are required because of the poor heat-conduction qualities of oil. Oil-miscible refrigerants are advantageous in that they give better lubrication because the refrigerant acts as a carrier of oil to the moving parts; they also eliminate oil-separation problems and aid in the return of oil from the evaporator.

Toxicity. The toxicity of a refrigerant may be of prime or secondary importance, depending upon the application. In comfort air conditioning

the danger of large volumes of a highly toxic refrigerant being discharged into the conditioned space must be reduced to a minimum. On the other hand, with small individual domestic refrigerators highly toxic refrigerants have been used satisfactorily and with no danger because of the small refrigerant charge. Table 5.14 gives the relative toxicity of the common refrigerants, based upon the concentration and the exposure time required for serious injury or death. The Underwriters' Laboratories class numbers indicate decreasing toxicity with increasing number.

TABLE 5.14
TOXIC PROPERTIES OF REFRIGERANTS⁴

Refrigerant	Kills or Seriously Injures			Poisonous Decomposition Products	Underwriters' Laboratories' Class No.
	Duration of Exposure	Concentration in Air			
		Per Cent by Volume	Pounds per 1000 cu ft at 70 F		
Sulfur dioxide.....	5 min.	0.7	1.2	...	1
Ammonia.....	½ hr.	0.5 to 0.6	0.2 to 0.3	...	2
Methyl formate.....	1 hr.	2 to 2.5	3.1 to 3.9	...	3
Methyl chloride.....	2 hr.	2 to 2.5	2.8 to 3.3	Yes	4
Dichloroethylene (Dielene)	2 hr.	2 to 2.5	5.0 to 6.3	Yes	4
Ethyl chloride.....	1 hr.	4.0	6.7	Yes	4-5
Methylene chloride (Carrene-1).....	½ hr.	5.1 to 5.3	11.2 to 11.70	Yes	4-5
Trichlorotrifluoroethane (Freon-113).....	1 hr.	4.8 to 5.3	23.3 to 25.2	Yes	4-5
Dichloromonofluoromethane (Freon-21).....	½ hr.	10.2	27.1	Yes	4-5
Carbon dioxide.....	½ to 1 hr.	29.0 to 30.0	33.0 to 34.2	No	5
Trichloromonofluoromethane (Freon-11, Carrene-2)	2 hr.	10	35.7	Yes	5
Monochlorodifluoromethane (Freon-22).....	2 hr.	18.0 to 22.6	40.2 to 50.5	Yes	5A
Dichlorodifluoromethane (Freon-12).....	2 hr.	28.5 to 30.4	88.5 to 94.4	Yes	6

In addition to the actual toxic effects of a refrigerant, consideration should be given to the irritating effects, which may result in a panic hazard if leakage occurs into a space occupied by many people. Ammonia and sulfur dioxide, being strong irritants, would not be advisable in comfort air-conditioning applications even if they were nontoxic. In contrast, all the Freon refrigerants and carbon dioxide are odorless within the range of concentrations encountered from refrigerant leakage.

⁴ Compiled from Underwriters' Laboratories Reports Misc. Hazard 2375 (1933), Misc. Hazard 2630 (1935), Misc. Hazard 3072 (1941), and Misc. Hazard 3134 (1940).

The threshold of odor detection for ammonia is 53 parts per million by volume; 700 parts per million will cause irritation. Sulfur dioxide is detectable at 3 to 5 parts per million, and 20 parts per million will cause irritation. The Freon refrigerants are odorless up to concentrations less than 20 per cent by volume in air. One per cent acrolein is sometimes added to methyl chloride as a warning agent; this mixture is highly irritating at approximately 0.1 per cent by volume. Concentrations of carbon dioxide up to 3 per cent stimulate breathing, and those of approximately 5 per cent usually cause a sense of giddiness. There is no need to add a warning agent to the Freon refrigerants and carbon dioxide because it is almost impossible for them to reach fatal concentration through refrigerant leaks.

TABLE 5.15
EXPLOSIVE PROPERTIES OF REFRIGERANTS⁵

Refrigerant	Explosive Range, Concentration in Air		Maximum Explosion Pressure, psi	Time of Pressure Develop- ment, Seconds
	Per Cent by Volume	lb per 1000 cu ft		
Ammonia.....	16. to 25.	7.1 to 11.5	50	0.175
Methyl formate.....	4.5 to 20.0	7.02 to 31.2	96	0.026
Methyl chloride.....	8.1 to 17.2	10.6 to 22.6	69	0.110
Dichloroethylene (Dielene).....	5.6 to 11.4	14.1 to 28.7	76	0.095
Ethyl chloride.....	3.7 to 12.0	6.21 to 20.1	87	0.028
Sulfur dioxide.....		Nonflammable		
Methylene chloride (Carrene-1).....		Nonflammable		
Carbon dioxide.....		Nonflammable		
Trichloromonofluoromethane (Freon-11, Carrene-2).....		Nonflammable		
Dichlorodifluoromethane (Freon- 12).....		Nonflammable		
Dichloromonofluoromethane (Freon-21).....		Nonflammable		
Monochlorodifluoromethane (Freon-22).....		Nonflammable		
Trichlorotrifluoroethane (Freon- 113).....		Nonflammable		

Explosive Properties. An ideal refrigerant should present no danger of explosion or fire, either in combination with air or in association with lubricating oils. Table 5.15 gives the explosive properties of the various common refrigerants and indicates that the only completely satisfactory refrigerants from this standpoint are sulfur dioxide, methylene chloride,

⁵ Compiled from Underwriters' Laboratories Reports Misc. Hazard 2375 (1933), Misc. Hazard 2630 (1935), Misc. Hazard 3072 (1941) and Misc. Hazard 3134 (1940).

carbon dioxide, and the Freon group. All the others are explosive within certain concentration limits.

Effect on Perishable Materials. In case of leakage an ideal refrigerant should have no effect on any perishable materials with which it may come in contact. The effects of ammonia vapors on most textiles, fruits, and vegetables are not dangerous; however, excessive exposure of most produce to ammonia will result in burns and rot. Many foods when exposed to ammonia will become unpalatable.

Freon vapors have no effect upon dairy products, meats, vegetables, flowers, plant life, furs, and fabrics. Products exposed to Freon vapors show no change of taste, color, or texture.

Methyl chloride vapors have no effect upon furs, textiles, or flowers. There is no danger in eating foods or drinking beverages exposed to methyl chloride vapors.

Sulfur dioxide will kill flowers and plant life, but it is not dangerous to foods. It acts as a bleaching agent on some dyes but is noninjurious to furs and to textile fibers.

Leakage Tendency. It is desirable that the leakage tendency of a refrigerant be low. If it does leak, it should be easily detectable. The rate of leak detection is dependent upon many factors (see §5.12, *Cost*). In all cases, increased tendency toward leakage and difficulty in detecting leaks will increase operating costs. Ammonia leaks are usually detectable by odor and may be located by the ammonium sulfite (white fumes) formed in the presence of burning sulfur. The presence of oil on pipe fittings and other connections usually indicates leakage within any system.

Halides, such as the Freon group of refrigerants, may be detected by means of a halide leak detector (see Fig. 5.2) consisting of an alcohol torch so designed as to permit air from a particular point being drawn into contact with copper in the flame. Freon leaks of a rate as low as one ounce per month may be detected in this manner by the resulting blue-green flame. However, periodic testings must be made or the loss of such a nonodorless refrigerant may be appreciable before it is noticed. The external presence of oil should be taken as a warning signal, since oil usually leaks out with the Freon. Leakage of Freon refrigerants may also be detected by the addition of a small amount of oil-soluble dye to the oil circulated in the system.

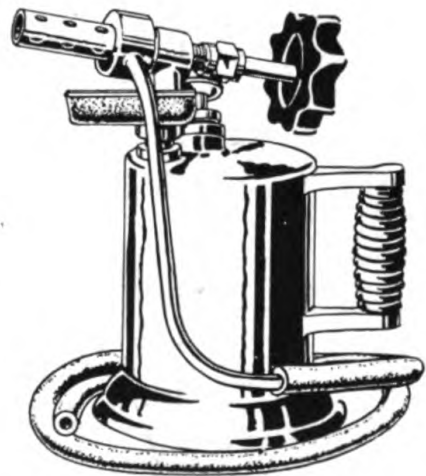


Fig. 5.2. A halide torch burns with a blue-green flame when halides are present.

Sulfur dioxide leaks are usually detectable by odor and may be located, as for ammonia, by the formation of ammonium sulfide. In this case aqueous ammonia solution is exposed to the leaking refrigerant.

Methyl chloride leaks are readily detectable by odor if acrolein has been added as a warning agent. Otherwise, as in the case of other comparatively odorless yet flammable refrigerants, leaks may be detected by placing the system under pressure and applying a solution of soap and water at the point of a suspected leak.

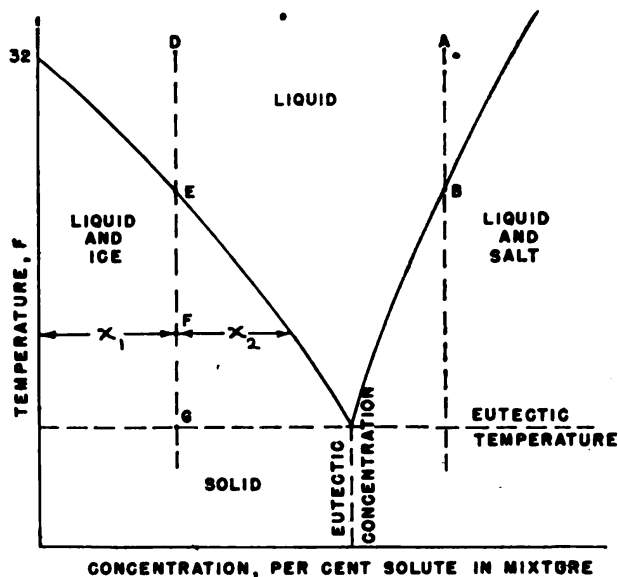


Fig. 5.3. Typical brine phase diagram.

5.13. Brines and Eutectics. *Brines* are frequently used as secondary refrigerants. In the broadest interpretation, they include not only salt solutions but also any liquids conveying heat from the source of the load to the evaporator. *Eutectic brines* consist of water and one or more additional ingredients proportioned to result in the mixture with the lowest freezing point. In addition to their application as secondary refrigerants, brines are also used to store refrigeration through the use of holdover tanks as well as for brine fogs or sprays for cooling meat, fish, and other foods. Eutectic brines are frequently used for refrigeration storage by freezing the mixture in metal containers or holdover plates. Such eutectic plates are used in some refrigerated trucks.

Most two-component brines have constitution diagrams indicating the relationship between temperature, phase, and concentration similar to that shown in Fig. 5.3. This diagram indicates that the mixture (if a salt brine is under consideration) may be liquid, liquid and salt, liquid and ice, or solid, depending upon the temperature and concentration. If the mixture has the eutectic concentration, no solidification occurs until the eutectic temperature is reached. At that temperature the entire mass solidifies as a mechanical mixture of ice and a dehydrate of the salt.

If the mixture has a solute concentration greater than that of the eutectic mixture (such as point *A*), with a lowering of the temperature salt starts to freeze from the solution when point *B* is reached. With further lowering of the temperature, salt continues to solidify, and the remaining liquid decreases in concentration, following the boundary line, until the eutectic concentration and temperature are reached. With a further drop in temperature the remaining mass freezes simultaneously.

If the mixture has a solute concentration lower than that of the eutectic mixture, pure ice instead of salt freezes from the solution until the eutectic temperature and concentration is reached. The path *D-E-F-G* represents the progress of such a mixture when the temperature is thus lowered. When the mixture is in the liquid-and-ice or liquid-and-salt regions, the proportions of each phase at any temperature may be determined by drawing a horizontal line through the state point intersecting the adjacent boundaries. The linear distances from these boundaries are inversely proportional to the phase percentages. Thus at point *F* the percentage of ice is $\frac{x_2}{x_1 + x_2}$, and the percentage of liquid is $\frac{x_1}{x_1 + x_2}$.

In determining the amount of heat absorbed or rejected in the thawing or freezing of brines, correction must be made for the heat of solution.

TABLE 5.16
AQUEOUS SOLUTION EUTECTIC POINTS⁶

Solute	Eutectic Concentration, Wt Solute Wt Solution	Eutectic Temperature, F
Ammonium chloride.....	0.191	+ 4.3
Calcium chloride.....	0.324	- 59.8
Sodium chloride.....	0.233	- 6.0
Sodium sulfate.....	0.049	+ 30.0
Potassium chloride.....	0.197	+ 12.8
Potassium nitrate.....	0.112	+ 26.6
Methyl alcohol.....	0.695	-164
Ethyl alcohol.....	0.763	- 94
Glycerine.....	0.669	- 60.4

Eutectic concentrations and temperatures for several combinations of materials are presented in Table 5.16. The total heats for sodium chloride and calcium chloride solutions at various concentrations and temperatures are given in Appendix Tables A.11 and A.12.

⁶ Adapted from *Refrigerating Data Book*, 5th ed., 1943, Chapter 6, Table 11.

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PROBLEMS

5.1. A direct-expansion evaporator consists of 45 ft of $\frac{3}{8}$ -in. I.D. finned tubing. If the coil is operating 50 per cent filled with liquid methyl chloride at 20 F, determine (a) the weight of liquid refrigerant in the coil and (b) the weight of refrigerant vapor. If the coil is evacuated by pumping a vacuum of 25 in. of mercury, determine (c) the weight of refrigerant vapor remaining in the evaporator.

5.2. A cylinder of Freon-22 is stored in a room at a temperature of 100 F. What pressure exists in the cylinder when (a) the valve is closed, and (b) when the valve is opened, allowing vapor to escape?

5.3. What must be the rate of refrigerant flow through a 10-ton ammonia system, operating with dry compression and neither liquid subcooling nor superheating of the evaporator vapor, when the suction gage pressure is 15.72 psi and the condenser gage pressure is 151.7 psi?

5.4. From data in the methyl chloride tables, construct a pressure-enthalpy chart on graph paper. Show both the liquid and vapor saturation lines and several constant-temperature lines extending into the superheated region.

5.5. A space 100×50 ft with a 20-ft ceiling is to be cooled. Determine the number of pounds of refrigerant which, when released in the space, will create a dangerously toxic atmosphere if the refrigerant is (a) Freon-12, (b) ammonia. (c) Determine the number of pounds of these refrigerants that will result in explosive concentrations in a compressor room $20 \times 10 \times 12$ ft.

5.6. The sodium chloride brine in a hold-over system is to be cooled to 16 F and will rise in temperature to 38 F when its accumulated capacity is being used. If the brine is to be maintained in the liquid state at all times and the heat gain to the storage tank is negligible, determine the weight and volume of brine required to supply a cooling load of 30,000 Btu in 3 hr when (a) the compressor is not operating, and (b) the compressor (which is of sufficient capacity to cool the holdover system from 38 F to 16 F in 12 hr) is in operation.

5.7. From the data on calcium chloride brine solutions in the Appendix, construct on graph paper a temperature-concentration diagram showing the boundaries between the liquid, liquid-and-ice, and solid regions. Indicate eutectic concentration and temperature.

5.8. A 10 per cent calcium chloride brine is lowered to a temperature of -10 F. From Problem 5.7 determine (a) the percentages of liquid and ice in the mixture

and (b) the concentration of the remaining liquid. If the mixture is lowered to the eutectic temperature, determine (c) the percentages of eutectic and of ice in the mixture.

5.9. Sodium chloride brine of specific gravity 1.18 is used in eutectic storage plates to counteract a heat leakage of 560 Btu per hour into a portable frozen-food storage chest. The plates are periodically removed from the chest and refrozen. If the brine temperature must not rise above -2°F , determine (a) the pounds of eutectic solid that must be supplied for a storage period of 24 hr and (b) the volume of eutectic solution that must be sealed in the plates.

CHAPTER 6

Air-cycle, Ejector, and Centrifugal Refrigeration, and the Heat Pump

6.1. Introduction. In addition to vapor-compression refrigeration, accomplished by reciprocating or rotary compressors, there are several additional types of systems. Air-cycle refrigeration, one of the earliest forms of cooling developed, became obsolete for many years because of its low coefficient of performance and high operating costs. It has, however, been applied recently to aircraft refrigerating systems, where with low equipment weight it can utilize a portion of the cabin-air supercharger capacity. It is unique in that the refrigerant remains in the gaseous phase throughout the cycle.

Steam-jet refrigeration has received limited usage in air conditioning and other high-temperature applications above the freezing point of water. Its thermodynamic cycle is essentially the same as for the standard vapor-compression system, but with a steam jet used for evacuation of the cooler and for compression of the refrigerant, water vapor. The ejector cycle may also be used with refrigerants other than water.

Centrifugal refrigeration employs the standard vapor-compression cycle, with compression accomplished by centrifugal means. Until recently its application was limited to comparatively large systems, but new developments permit its use in sizes down to 50 tons and possibly much lower. Absorption refrigeration systems, which accomplish compression by absorbing the refrigerant leaving the evaporator in a solution at low temperature and pressure and then distilling off the refrigerant at a high temperature and pressure by the application of heat, are discussed in Chapter 15.

The heat pump, using the standard refrigeration cycle to pump heat from some external, comparatively low-temperature source to the inside space to be warmed, has been developed to the point of practical application in the last few years. Here the useful energy is the heat discharged from the condenser instead of the heat absorbed by the evaporator. With the exception of absorption refrigeration, the theory involved in each of these systems will be discussed in this chapter.

6.2. Air-Cycle Refrigeration. The air-cycle refrigeration system is the only air-cooling process developed for wide commercial application in which a gaseous refrigerant is used throughout the cycle. Compression is accomplished by a reciprocating or centrifugal compressor as in the

vapor-compression cycle, but condensation and evaporation are, of necessity, replaced by a sensible cooling and heating of the gas. An *air cooler* is used in place of a condenser and a *refrigerator* in place of an evaporator. Although some cooling of the gas would result from its expansion, irreversibly, through a throttling valve (see §16.4), a greater drop in air temperature is accomplished if the expansion is controlled to approach the isentropic. Therefore the expansion valve is replaced by an *expansion engine* or turbine.

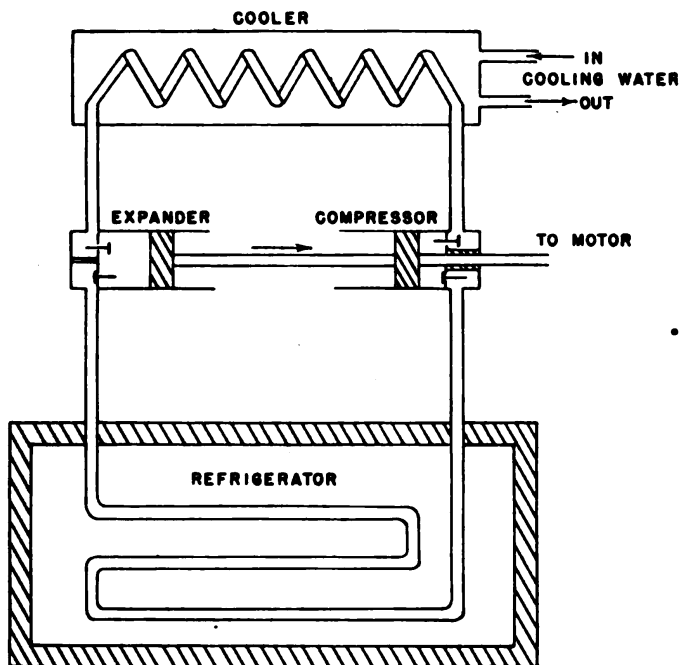


Fig. 6.1. Air refrigeration system.

Since constant-temperature condensation and evaporation are no longer involved, it becomes impractical to attempt to approach the reversed Carnot cycle, and the reversed Brayton cycle is substituted. Figure 6.1 shows diagrammatically an air-cycle refrigeration system with air compressor, cooler, expander, and refrigerator. Here the work gained in the expander is used to aid in compression of the air and thus in decreasing the net external work required for operation of the system. In practice this may or may not be done. For example, in some aircraft refrigerating systems of the air-cycle type, the expansion work may be dissipated efficiently by using it to drive other devices. In the theoretical analysis to follow, the work of expansion is assumed to aid air compression.

Figure 6.2 shows theoretical p - V and T - s diagrams for the reversed Brayton cycle. Here it is assumed that absorption and rejection of heat are constant-pressure processes and that compression and expansion are isentropic processes. Thus the heat absorbed in the refrigerator is

$${}_2Q_3 = Wc_p(T_3 - T_2) \quad (6.1)$$

and the heat rejected in the cooler is

$${}_4Q_1 = Wc_p(T_4 - T_1) \quad (6.2)$$

where Q denotes total quantity of heat, W weight, c_p specific heat at constant pressure, and T temperature. The steady-flow work of compression, if the process is considered to be polytropic, will be

$${}_3W_{k_4} = \frac{n}{n-1} (p_4V_4 - p_3V_3) \quad (6.3)$$

and expansion will be

$${}_1W_{k_2} = \frac{n}{n-1} (p_1V_1 - p_2V_2) \quad (6.4)$$

where W_k denotes work, n the expansion exponent, p pressure, and V volume. Equations 6.3 and 6.4 may easily be reduced to the theoret-

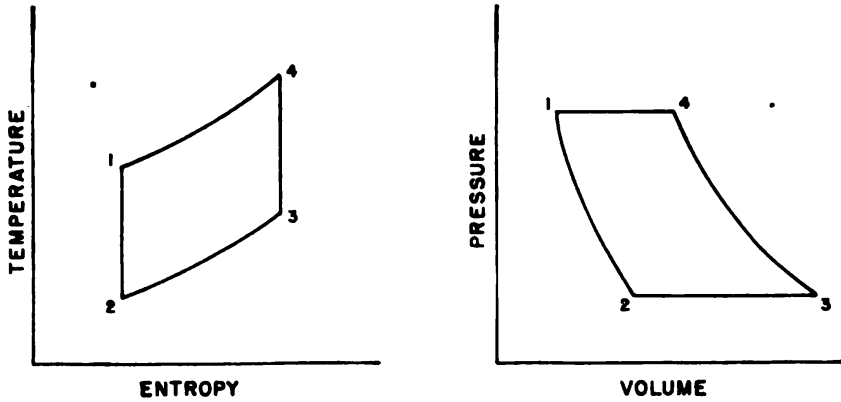


Fig. 6.2. Reversed Brayton cycle.

ical isentropic processes shown in Fig. 6.2 by the substitution of $\gamma = n$ and the known relationship

$$R = c_p \left(\frac{\gamma - 1}{\gamma} \right) J \quad (6.5)$$

The net external work required for operation of the cycle is

$$\begin{aligned} {}_3W_{k_4} - {}_1W_{k_2} &= \frac{n}{n-1} (p_4V_4 - p_3V_3 - p_1V_1 + p_2V_2) \\ &= \left(\frac{n}{n-1} \right) \left(\frac{\gamma - 1}{\gamma} \right) Wc_p(T_4 - T_3 + T_2 - T_1) \end{aligned} \quad (6.6)$$

or, for isentropic compression and expansion,

$$W_{k_{\text{net}}} = Wc_p(T_4 - T_3 + T_2 - T_1) \quad (6.7)$$

This expression is identical with the net heat rejected from the cycle, since by the law of conservation of energy, the net work on the gas must be equivalent to the net rejected heat.

From Chapter 4 the coefficient of performance is defined as

$$\text{c.p.} = \frac{Q_{\text{added}}}{Q_{\text{rejected}} - Q_{\text{added}}} = \frac{Q_{\text{added}}}{W_{\text{knet}} \text{ (in Btu)}}$$

For the air cycle, assuming polytropic compression and expansion, the coefficient of performance is

$$\begin{aligned} \text{c.p.} &= \frac{W c_p (T_3 - T_2)}{\left(\frac{n}{n-1}\right) \left(\frac{\gamma-1}{\gamma}\right) W c_p (T_4 - T_3 + T_2 - T_1)} \\ &= \frac{T_3 - T_2}{\left(\frac{n}{n-1}\right) \left(\frac{\gamma-1}{\gamma}\right) (T_4 - T_3 + T_2 - T_1)} \end{aligned} \quad (6.8)$$

and for isentropic compression and expansion,

$$\text{c.p.} = \frac{T_3 - T_2}{T_4 - T_3 + T_2 - T_1} \quad (6.9)$$

In the actual air-refrigeration cycle isentropic expansion will result in the greatest temperature drop and, therefore, an attempt is made to approach this process as closely as possible by insulating the expander from the surrounding air. If possible, isothermal compression of the air would be desirable, since it would reduce the required work. Water-jacketing of the compressor cylinders will therefore reduce somewhat the work of compression and at the same time will aid lubrication by reducing the average temperature during compression. Although cylinder cooling by normal means will aid in this respect, the resulting process is polytropic and by no means approaches the isothermal. If polytropic compression and isentropic expansion are assumed, the coefficient of performance is

$$\text{c.p.} = \frac{T_3 - T_2}{\left(\frac{n}{n-1}\right) \left(\frac{\gamma-1}{\gamma}\right) (T_4 - T_3) + (T_2 - T_1)} \quad (6.10)$$

Air-cycle refrigeration may be designed and operated either as an open or a closed system. In the *closed* or *dense-air system* the air refrigerant is contained within the piping or component parts of the system at all times and with the refrigerator usually maintained at pressures above atmospheric. The term "dense-air system" is derived from the higher pressures maintained in comparison with the open system. In the *open system* the refrigerator is replaced by the actual space to be cooled with the air expanded to atmospheric pressure, circulated through the cold room and then compressed to the cooler pressure. Such a system is inherently limited to operation at atmospheric pressure in the refrigerator. It has the additional disadvantage that moisture is continually being

picked up by the air circulated through the cooled space and then deposited as frost or ice at the lowest temperature points of the system, usually at the expansion-engine discharge.

The open and dense-air systems, as originally applied to cold-storage and theater cooling systems, have become almost obsolete. The original advantage possessed by the systems of using a completely safe and cheap refrigerant has been nullified by the introduction of such nontoxic modern refrigerants as the Freon group, and it has been impossible to overcome the inherently high horsepower required per ton of refrigeration. However, the air-cycle system is basically well adapted to aircraft refrigeration. This modern application is discussed in §7.10.

EXAMPLE 6.1. An air refrigeration system operating on an open air cycle is required to produce 20 tons of refrigeration with a cooler pressure of 175 psia and a refrigerator pressure of 15 psia. The temperature of the air leaving the cooler is 70 F, and the air leaving the room is at 30 F. Assuming a theoretical cycle with isentropic compression and expansion, with no compressor clearance and no losses, determine (a) weight of air circulated per minute, (b) compressor piston displacement required per minute, (c) expander displacement required per minute, (d) coefficient of performance, and (e) horsepower required per ton of refrigeration.

SOLUTION:

$$\begin{aligned} (a) \quad T_1 &= 70 \text{ F} = 530 \text{ F abs} \\ T_2 &= 530 \left(\frac{175}{15} \right)^{\frac{1.41-1}{1.41}} = 260 \text{ F abs} \\ T_3 &= 30 \text{ F} = 490 \text{ F abs} \\ T_4 &= 490 \left(\frac{175}{15} \right)^{\frac{1.41-1}{1.41}} = 1000 \text{ F abs} \end{aligned}$$

Refrigerating effect per lb air circulated = $0.24(490 - 260) = 55.2$ Btu per lb

$$\text{Weight air circulated per min} = \frac{200 \times 20}{55.2} = 72.4 \text{ lb}$$

$$\begin{aligned} (b) \quad \text{Compressor piston displacement} &= V = \frac{WRT}{p} \\ &= \frac{(72.4)(53.3)(490)}{(15)(144)} \\ &= 875 \text{ cu ft per min} \end{aligned}$$

$$\begin{aligned} (c) \quad \text{Expander displacement} &= V_2 = V_3 \frac{T_2}{T_3} \\ &= 875 \left(\frac{260}{490} \right) = 464 \text{ cu ft per min} \end{aligned}$$

$$(d) \text{ c.p.} = \frac{T_3 - T_2}{T_4 - T_3 + T_2 - T_1} = \frac{490 - 260}{1000 - 490 + 260 - 530} = 0.96.$$

$$\begin{aligned} (e) \quad \text{Total horsepower} &= \frac{Wc_p(T_4 - T_3 + T_2 - T_1)}{42.42} \\ &= \frac{(72.4)(0.24)(240)}{42.42} = 98.3 \end{aligned}$$

$$\text{Horsepower per ton} = \frac{98.3}{20} = 4.92$$

EXAMPLE 6.2. The air-refrigeration system of Example 6.1 is modified to operate as a closed system with the pressure in both the cooler and the refrigerator increased by 25 psi. The same compressor is used, but the expander and the other parts of the system are changed to meet the requirements of the new design. The temperatures leaving the cooler and the refrigerator are still maintained at 70 F and 30 F, respectively. Determine (a) weight of air circulated per minute, (b) tons of refrigeration developed by the system, (c) expander displacement required per minute, (d) coefficient of performance, and (e) horsepower required per ton of refrigeration.

SOLUTION:

$$\begin{aligned} \text{(a)} \quad T_1 &= 530 \text{ F abs} \\ T_2 &= 530\left(\frac{40}{200}\right)^{0.291} = 332 \text{ F abs} \\ T_3 &= 490 \text{ F abs} \\ T_4 &= 490\left(\frac{200}{40}\right)^{0.291} = 782 \text{ F abs} \end{aligned}$$

$$\begin{aligned} \text{Weight air circulated per min} = W &= \frac{pV}{RT} \\ &= \frac{(40)(144)(875)}{(53.3)(490)} = 193 \text{ lb} \end{aligned}$$

$$\begin{aligned} \text{(b)} \quad \text{Refrigerating effect per lb air circulated} &= 0.24 (490 - 332) \\ &= 37.9 \text{ Btu} \end{aligned}$$

$$\text{Tons refrigeration developed} = \frac{37.9 \times 193}{200} = 36.6$$

$$\text{(c)} \quad \text{Expander displacement} = 875\left(\frac{332}{782}\right) = 593 \text{ cu ft per min}$$

$$\text{(d)} \quad \text{c.p.} = \frac{490 - 332}{782 - 490 + 332 - 530} = 1.68$$

$$\text{(e)} \quad \text{Total horsepower} = \frac{(19.3)(0.24)(782 - 490 + 332 - 530)}{42.42} = 102.8$$

$$\text{Horsepower per ton} = \frac{102.8}{36.6} = 2.81$$

6.3. Ejector-Cycle Refrigeration. Any refrigerant may be placed in a flash chamber and caused to produce refrigeration at the desired temperature by evacuating the space over the liquid refrigerant to the corresponding pressure. If the volume of vapor formed by the evaporating refrigerant is comparatively small, a *reciprocating* or *rotary compressor* may be used for its removal. If the volume of gas to be compressed is large, some other means of compression must be adopted, since the size of a reciprocating or rotary unit required would make its application prohibitive. Both *ejector* and *centrifugal* systems are satisfactory where the volume of vapor to be compressed is high.

Whereas any refrigerant may be made to operate in an ejector system, although not always efficiently, water vapor has several inherent advantages for this type of application. In some cases steam-generating equipment is already available for use with the refrigerating system, so that additional auxiliary equipment is not necessary. Also, an excess supply

of high-pressure steam may be available for economical use as the motive force in the ejector, in which case no heat exchanger is necessary in the evaporator, since the water may be circulated directly as a secondary refrigerating medium. Moreover, water vapor is nontoxic, nonexplosive, and cheap. Its disadvantages are that it is limited to usage above freezing temperatures, that it requires extremely low evaporator pressures, and that it needs compression equipment of very high volumetric capacity. In fact, ejector and centrifugal systems are the only ones to date that have been found at all satisfactory for use with water vapor. With the exception of a few experimental installations almost the entire devel-

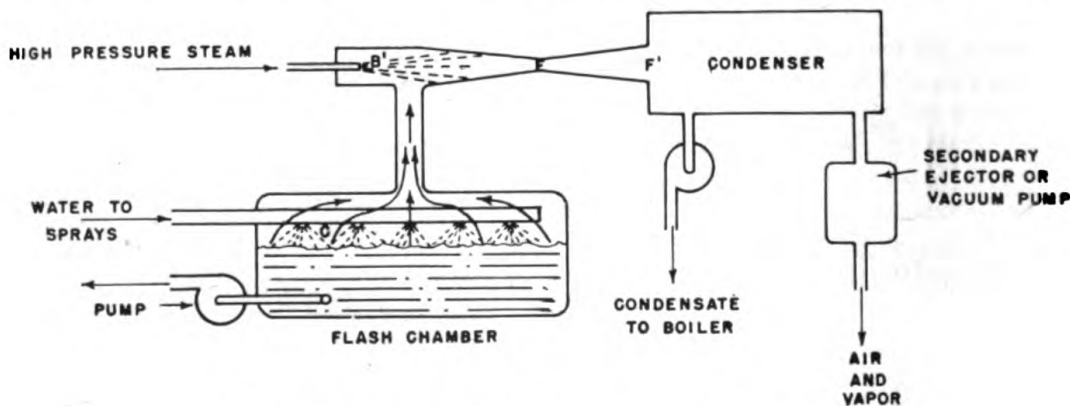


Fig. 6.3. Typical steam-jet water-vapor refrigerating system.

opment of the ejector system has been with water vapor or steam as the refrigerant as well as the motive force.

6.4. Steam-Jet Water-Vapor System. A typical *steam-jet* water-vapor refrigerating system is shown diagrammatically in Fig. 6.3, and the corresponding temperature-entropy diagram is shown in Fig. 6.4. High-pressure steam is supplied to the nozzle inlet and is expanded to B' in the combining chamber. Here water vapor, originating from the flash chamber at C , is entrained with the high-velocity steam jet and compressed through E into the diffuser at F' . The velocity of the mixture is then reduced, the kinetic energy is converted into an increase in static head, and the mass is discharged to the condenser. The condensate from the condenser is usually returned to the boiler, and any water vapor flashed from the evaporator and included in this condensate must be replaced by make-up water added to the cold-water circulating systems. This substitution is usually accomplished by a float valve that allows the addition of sufficient water to maintain a constant water level in the flash chamber. The water in flash chamber C is cooled by the absorption from it of the latent heat of that portion of water flashed and removed by the ejector. The amount of water lost in the cooling process is very small, and the remaining water is circulated from the chamber as a secondary cooling medium to the point of the load. Upon return it is usually

sprayed into the flash chamber to provide exposure of the maximum surface area.

The operating pressures required are extremely low, since an evaporation temperature of approximately 40 F requires an absolute pressure in the evaporator of 0.25 in. of mercury. If condensation is to be accomplished at approximately 100 F, the condenser pressure must be 2 in. of mercury absolute. Operation under these conditions results in a compression ratio of 8 to 1, which approaches the limit for efficient operation. The air and vapor remaining in the condenser are removed by means of

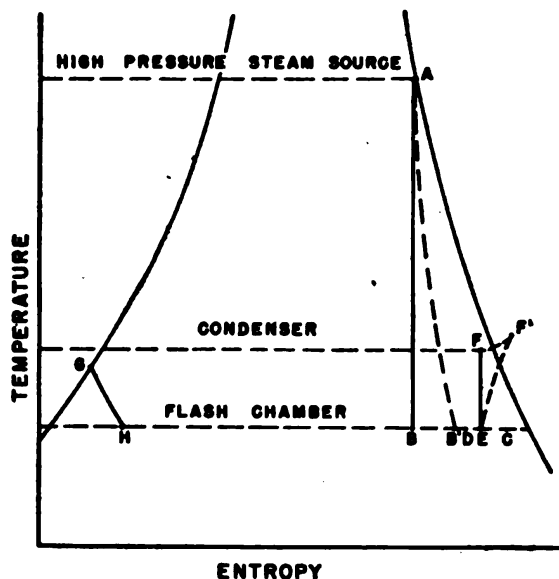


Fig. 6.4. Steam-jet refrigeration cycle.

a secondary ejector or a vacuum pump and are discharged from the system. If temperatures below 32 F are to be maintained in the evaporator, an antifreeze may be added or a brine may be used. However, the use of water-vapor refrigeration below freezing is uncommon because, among other factors, of the extremely high vacuums that must be maintained.

In Fig. 6.4 point *A* represents the initial condition of the motive steam before passage through the nozzle and *B* the final condition, assuming isentropic expansion. Point *C* represents the initial condition of the water vapor flashed in the evaporator and *E* the resultant condition of the mixture of high-velocity steam and entrained water vapor before compression. If compression were isentropic, the final condition would then be *F* at discharge of the mixture to the condenser. Make-up water is available at *G*, slightly lower in temperature than the condenser conditions, and is throttled to point *H* in the flash chamber. Because of unavoidable losses encountered in expansion, entrainment, and compression, the actual path of expansion and entrainment of the motive steam is *AB'D*. The motive steam at *D* and the water vapor flashed at

C mix to give a resultant condition shown as E . Compression of the mixture actually occurs along EF' .

The nozzle efficiency, η_n , is the ratio of the actual enthalpy drop to the isentropic enthalpy drop of the motive steam passing through the nozzle. Numerically, this may range from 85 to 90 per cent. The entrainment efficiency, η_e , indicates the losses that originate in the transfer of a portion of the kinetic energy of the motive steam to the flashed vapor. This process, in which the flashed vapor is brought into motion at the expense of a loss in kinetic energy of the motive steam, is inherently inefficient and, according to Kalustian,¹ should be taken as 65 per cent. The loss of this kinetic energy, $1 - \eta_e$, is dissipated as heat added to the mixture. The compression efficiency, η_c , indicates the loss encountered in utilizing the kinetic energy of the mixture for compression in the diffuser. The ratio between the actual enthalpy increase needed to accomplish this compression to the isentropic enthalpy increase theoretically required is the compression efficiency, which may be taken as 75 to 80 per cent.

By definition of nozzle efficiency,

$$h_A - h_{B'} = \eta_n(h_A - h_B) \quad (6.11)$$

The quantity $h_A - h_{B'}$ is equivalent to the kinetic energy of the motive steam, now available for entrainment of the vapors in the flash chamber. This process, however, is an inefficient one, and the portion of the original motive force available for compression to the condenser is

$$h_A - h_D = \eta_e(h_A - h_{B'}) \quad (6.12)$$

However, the actual energy required for compression of the mixture is

$$h_{F'} - h_E = \frac{h_F - h_E}{\eta_c} \quad (6.13)$$

Since the required work of compression must be equal to the available energy for compression, the members of equations 6.12 and 6.13 must be equal when adjusted for the weights involved. Thus

$$\begin{aligned} (W + 1)(h_{F'} - h_E) &= W(h_A - h_D) \\ (W + 1) \frac{h_F - h_E}{\eta_c} &= W\eta_n\eta_e(h_A - h_B) \end{aligned}$$

$$\text{and} \quad W = \frac{h_F - h_E}{(h_A - h_B)\eta_n\eta_e\eta_c - (h_F - h_E)} \quad (6.14)$$

where W is the weight of motive steam in pounds per pound of flashed vapor. For each pound of vapor flashed from the evaporator,

$$\text{Refrigerating effect} = h_c - h_g \text{ Btu per lb} \quad (6.15)$$

$$\text{and} \quad \text{Weight motive steam per (ton)(hr)} = \frac{12,000W}{h_c - h_g} \quad (6.16)$$

¹ Kalustian, Peter, "Analysis of the Ejector Cycle." *Refrigerating Engineering*, Vol. 28, No. 4 (October, 1934), p. 188.

This expression serves as a practical means of evaluating the efficiency of a steam-jet system, since, as in the absorption system, little mechanical energy is involved, and the term "coefficient of performance" has no true meaning.

EXAMPLE 6.3. A steam-ejector water-vapor system is supplied with motive steam at 100 psia saturated, with the water in the flash chamber at 40 F. Make-up water is supplied to the cooling system at 65 F, and the condenser is operated at 2 in. of mercury absolute. The nozzle efficiency is 88 per cent, the entrainment efficiency 65 per cent, and the compression efficiency 80 per cent. The quality of the motive steam and flashed vapor at beginning of compression is 92 per cent. Determine (a) weight of motive steam required per pound of flash vapor, (b) quality of vapor flashed from flash chamber, (c) refrigerating effect per pound of flash vapor, (d) weight of motive steam required per (hour)(ton of refrigeration), and (e) volume of vapor removed from flash chamber per (hour)(ton of refrigeration).

SOLUTION:

(a) Referring to Fig. 6.4,

$$x_B = \frac{1.6026 - 0.0162}{2.1597 - 0.0162} = 0.74$$

$$h_B = 8.05 + 0.74(1079.3 - 8.05) = 800.8 \text{ Btu per lb}$$

By equation 6.11,

$$h_A - h_{B'} = 0.88(1187.2 - 800.8) = 340.0$$

$$h_{B'} = 847.2 \text{ Btu per lb}$$

$$x_{B'} = \frac{847.2 - 8.05}{1079.3 - 8.05} = 0.783$$

By equation 6.12,

$$h_A - h_D = (0.65)(340.0) = 221.0$$

$$h_D = 966.2$$

$$x_D = \frac{966.2 - 8.05}{1079.3 - 8.05} = 0.895$$

At point E,

$$x_E = 0.92$$

$$h_E = 8.05 + 0.92(1079.3 - 8.05) = 993.6$$

$$s_E = 0.0162 + 0.92(2.1597 - 0.0162) = 1.9720$$

By equation 6.13,

$$s_F = s_E = 1.9720$$

$$x_F = \frac{1.9720 - 0.1316}{1.9797 - 0.1316} = 0.997$$

$$h_F = 69.10 + 0.997(1105.7 - 69.10) = 1102.6$$

$$h_{F'} = 993.6 + \frac{1102.6 - 993.6}{0.80} = 1129.8$$

By equation 6.14,

$$\begin{aligned} W &= \frac{1102.6 - 993.6}{(1187.2 - 800.8)(0.88)(0.65)(0.80) - (1102.6 - 993.6)} \\ &= 1.62 \text{ lb motive steam per lb flashed vapor} \end{aligned}$$

$$\begin{aligned}
 (b) \quad & h_C + Wh_D = (W + 1)h_E \\
 & h_C = (1.62 + 1)(993.6) - (1.62)(966.2) \\
 & \quad = 1038.0 \text{ Btu per lb} \\
 & x_C = \frac{1038.0 - 8.05}{1079.3 - 8.05} = 0.96
 \end{aligned}$$

(c) By equation 6.15,

$$\begin{aligned}
 \text{Refrigerating effect} &= 1038.0 - 33.05 \\
 &= 1004.9 \text{ Btu per lb flashed vapor}
 \end{aligned}$$

(d) By equation 6.16,

$$\frac{(12000)(1.62)}{1004.9} = 19.3 \text{ (lb motive steam) per (hr)(ton)}$$

$$\begin{aligned}
 (e) \quad & v_E = 0.0162 + 0.92(2444 - 0.0162) \\
 & \quad = 2248 \text{ cu ft per lb}
 \end{aligned}$$

Volume vapor removed from flash chamber

$$= \frac{(12000)(2248)}{1004.9} = 26800 \text{ (cu ft) per (hr)(ton)}$$

6.5. Centrifugal Refrigeration. The *centrifugal refrigeration cycle* is identical with the reciprocating or rotary compression cycle, but the means used to accomplish compression is basically different. In place of a positive means of compression the centrifugal compressor depends upon self-compression of the refrigerant by centrifugal force as well as by conversion of the kinetic energy of the high-velocity vapors into a static pressure increase. The desirable properties of a centrifugal-compressor refrigerant differ in several respects from those for reciprocating compression. With a reciprocating machine the displacement is fixed by the physical dimensions and the rpm of the compressor, and therefore it is desirable to use a refrigerant requiring a low volumetric displacement per ton of refrigeration developed. Centrifugal compressors, however, operate most efficiently with a refrigerant of high molecular weight and high volumetric displacement. Moreover, reciprocating compressors operate satisfactorily with refrigerants requiring a high compression ratio, whereas centrifugal compressors require a moderately low compression ratio. Thus a refrigerant such as Freon-11 is well suited for centrifugal compression (see Chapter 5).

Centrifugal compressors are sometimes adapted to water-vapor refrigeration with a centrifugal compressor supplanting the steam ejector. Water-vapor centrifugal compression, however, requires such high rotational speeds (10,000 rpm) for efficient operation that inherent mechanical difficulties have limited its application.

Practically all centrifugal compressors are designed with several stages of compression, thus requiring their classification as multistage. However, if intercoolers are not used between stages and if the weight of refrig-

erant compressed is the same for each stage, thermodynamically the process is equivalent to single-stage compression and may be so considered in calculations.

The forces involved in the operation of a centrifugal compressor may be best understood by reference to Fig. 6.5. Consider first a thin layer of gaseous refrigerant of area A , density ρ , thickness dx , and at a distance x from the axis. The impeller speed is ω radians per sec, the radius r ft, and the linear tip speed, v ft per sec. Then the volume of the element of refrigerant is $A dx$, and the centrifugal force F upon this element is

$$dF = \frac{\omega^2 A \rho x dx}{g}$$

if the outlet of the compressor is completely closed. The head, in feet of refrigerant, corresponding to this force is

$$dH = \frac{\omega^2 x dx}{g}$$

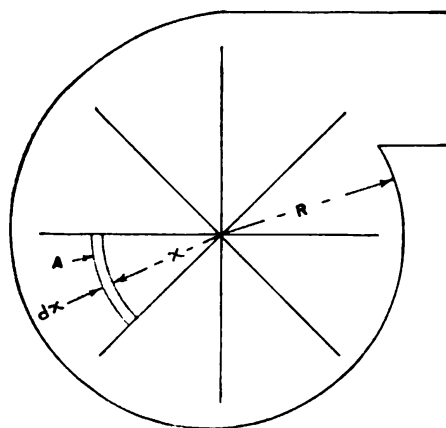


Fig. 6.5. Centrifugal compression forces.

and the total head developed from the axis to the outer radius r is

$$H_1 = \int_0^r \frac{\omega^2 x dx}{g} = \frac{\omega^2 r^2}{2g}$$

Since V is the linear tip speed and $V = \omega r$, the head developed by centrifugal force becomes

$$H_1 = \frac{V^2}{2g} \quad (6.17)$$

In addition to this force, produced centrifugally by passage of the refrigerant from the center of rotation to the periphery of the impeller, an additional head is produced by virtue of the mass being carried by the whirling impeller. As the mass of refrigerant attains a rotational speed approaching that of the rotor, this additional head in the form of kinetic energy, is

$$H_2 = \frac{V^2}{2g} \quad (6.18)$$

and the total head developed, assuming that the blades are radial without any curvature, is

$$\begin{aligned} H_T &= H_1 + H_2 \\ &= \frac{V^2}{g} \end{aligned} \quad (6.19)$$

This theoretical head may be converted to pressure increase by multiplying by the density and dividing by 144. Thus the pressure p , in pounds per square inch, is

$$p = \frac{V^2 \rho}{144g} \quad (6.20)$$

It is evident from equation 6.20 that the pressure developed is dependent upon the density of the refrigerant and the square of the impeller-tip velocity, and that a low-density refrigerant requires a much higher rotational speed than a high-density refrigerant if the same pressure increase is to be developed. Therefore a refrigerant such as Freon-11 with a density of 0.183 lb per cubic foot at 40 F will require lower compressor speeds and, in general, will be more satisfactory than water vapor with a density of 0.00041 lb per cubic foot at the same temperature.

In the design of centrifugal compressors the impeller blades may be radial, forward-curved, or backward-curved. Radial blades have received only limited commercial application, and forward-curved blades have been found unsatisfactory because of the compressors' unstable characteristics under conditions of varying load. Backward-curved blades have been found the most satisfactory, since the head developed falls off only slightly as the delivery increases, and therefore the compressor exhibits nonoverloading power characteristics with increasing refrigeration loads. Equations 6.19 and 6.20 are approached with radial blading but must be modified to take into account the effect of the discharge angle with either forward- or backward-curved blading.

The efficiency of a centrifugal compressor is usually expressed as the ratio between $n/(n - 1)$ for the actual polytropic process and $\gamma/(\gamma - 1)$ for the theoretical isentropic process. With modern centrifugal compressors the efficiency may range as high as 70 or 80 per cent at full load. The losses are primarily irreversible changes resulting from fluid friction and turbulence. The advantages of centrifugal compressors are simplified lubrication with the lubricant separated from the refrigerant, ease of capacity control and high efficiency at partial load, and concentration of a high capacity in a comparatively small machine. The principal disadvantages are the necessity for a large number of stages with refrigerants requiring a high compression ratio, and nonadaptability to small-tonnage equipment pumping small quantities of refrigerant.

6.6. The Heat Pump. In §4.2 it was shown that the Carnot cycle may, theoretically, be used as a heat engine or that it may be reversed and used either as a refrigerating machine or a heat pump. With the reversed Carnot cycle the coefficient of performance when used as a refrigerating machine is

$$\text{c.p.} = \frac{T_2}{T_1 - T_2} \quad (4.2)$$

and for a heat pump

$$\text{c.p.} = \frac{T_1}{T_1 - T_2} \quad (4.3)$$

With the refrigerating machine (equation 4.2) it is desired that heat be absorbed from the low-temperature source, and the quantity of this heat is proportional to the numerator, T_2 . With the heat pump (equation 4.3) it is desired that heat be rejected at the high-temperature sink, and the quantity of this heat is proportional to the numerator, T_1 . The difference lies only in the concept of the "desired effect." With an actual compression refrigeration machine operating as a heat pump, the coefficient of performance may be defined by the equation

$$\text{c.p.} = \frac{\text{heat rejected to condenser}}{\text{heat equivalent of net work of compression}} \quad (6.21)$$

The economical adaptation of the heat pump as a practical means of heating requires that the temperature of the source from which the heat is extracted be as high as possible and that the temperature of the sink to which it is rejected for heating purposes be as low as possible. The smaller the spread between evaporator and condenser temperatures, the lower is the required work of compression and the higher the coefficient of performance. Both water and air, therefore, are practical mediums to which the condenser heat may be rejected, but the generation of steam requires too high a condenser temperature. Economy therefore favors that the heat pump be used in conjunction with either water or air heating systems.

The source of heat supply for the evaporator may be air, ground water, or the earth itself, depending upon the climate and topography. As yet, utilization of solar radiation through low-temperature storage sumps has not been attempted, but apparently there is no reason why this might not provide a source of heat at or above 32 F with proper development. The problem of utilization of solar energy has never been one of the quantity of heat available but rather of its utility at comparatively low temperatures. Here is a possible application that fits these requirements.

Air is abundant, convenient, and easy to handle, but the heat-pump efficiency decreases and the heating requirements increase as the outside-air temperature drops. Also, defrosting becomes a problem at operating temperatures below freezing. If the air-temperature drop is kept low, large quantities of air must be handled. This would involve large equipment and possible noise problems. Thus practical considerations seem to limit the present systems using air to climates like those of the extreme southern parts of the United States, where temperatures under 20 F are not encountered.

Water from lakes, rivers, and wells may be used. Water from wells

is preferable since here the water temperature is fairly constant the entire year and is 50 F or higher even as far north as Minnesota. However, means for returning the water to the underground reservoir should be provided to prevent depletion should there be any widespread usage of such systems. A "water-using" unit would be smaller than an air unit, since higher temperatures are involved and since the volume of water is less than the volume of air that must be handled. Disadvantages of using well water include the problem of locating an adequate supply, the cost of the well, and the problem of water disposal.

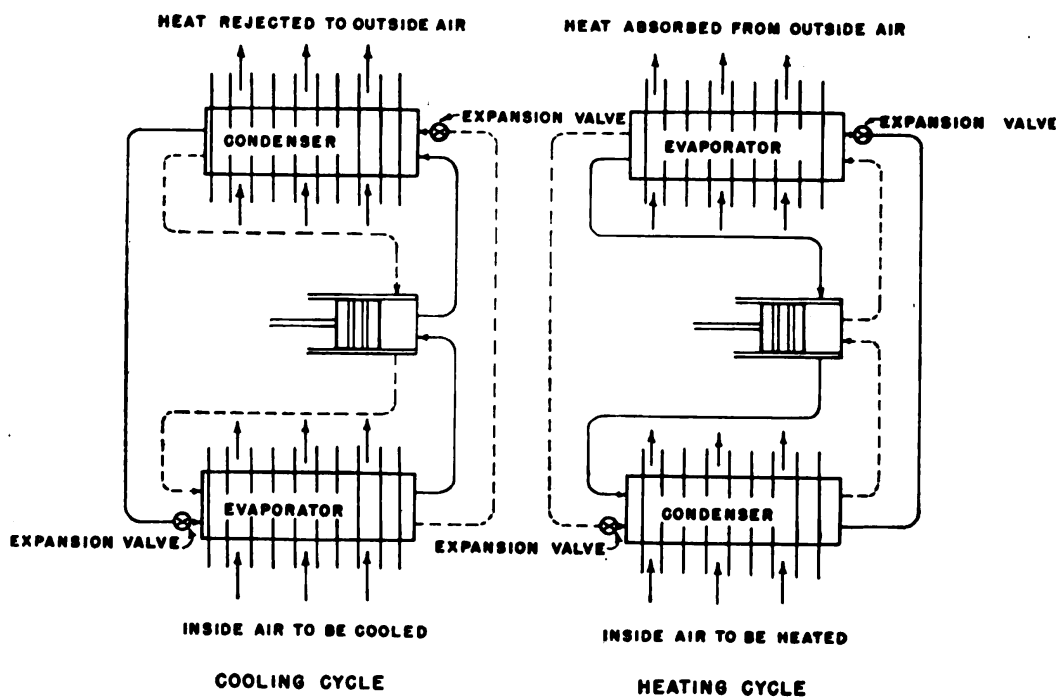


Fig. 6.6. Air-to-air cooling and heating loads.

Recently much consideration has been given to using the earth as a heat source.^{2,3} Installations have been made where the refrigerant coil was buried in the ground and also where a heat exchanger was used with a circulating-water coil buried in the ground. One disadvantage seems to be the large amount of heat-transfer surface required.

Since development of the heat pump to date includes the use of air, water, and earth as heat sources and air and water as heat sinks, there are six possible combinations of source and sink in application: air to air, air to water, water to air, water to water, earth to air, and earth to water. An air-to-air system is shown diagrammatically in Fig. 6.6, arranged for

² Kemler, E. N., "Heat for the Heat Pump." *Heating, Piping and Air Conditioning*, Vol. 18, No. 12 (December, 1946), p. 85.

³ Ambrose, Kemler, Smith, and Holladay, "Progress Report on the Heat Pump." *Heating and Ventilating*, Vol. 43, No. 12 (December, 1946), pp. 68-82.

both the heating cycle and the cooling cycle. Further application of the heat pump is discussed in Chapter 17.

EXAMPLE 6.3. A vapor-compression refrigeration system has a rated capacity of 10 tons of refrigeration when operating with Freon-12 isentropically between evaporator and condenser temperatures of 32 and 95 F, respectively. If there is no subcooling of the refrigerant leaving the condenser and no superheating of the refrigerant entering the compressor, determine (a) the coefficient of performance of the refrigerating machine, (b) the coefficient of performance of the system operating under the same conditions as a heat pump, and (c) the heating capacity of the system operating as a heat pump.

SOLUTION:

$$(a) \quad \text{c.p.} = \frac{81.8 - 29.5}{89.4 - 81.8} = 6.9$$

$$(b) \quad \text{c.p.} = \frac{89.4 - 29.5}{89.4 - 81.8} = 7.98$$

$$(c) \quad \text{Heating capacity} = (10)(200)(60) \frac{89.4 - 29.5}{81.8 - 29.5} \\ = 137,300 \text{ Btu per hr}$$

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PROBLEMS

6.1. An air refrigeration system, operating on a closed cycle, is required to produce 20 tons of refrigeration with a cooler pressure of 225 psia and a refrigerator pressure of 65 psia. Leaving-air temperatures are 70 F for cooler and 30 F for refrigerator. Assuming a theoretical cycle with isentropic compression and expansion, no compressor clearance and no losses, determine (a) pounds per minute of air circulated, (b) cubic feet per minute of compressor piston displacement, (c) cubic feet per minute of expander displacement, (d) coefficient of performance, and (e) net horsepower per ton required.

6.2. Solve Problem 6.1 if compressor and expander clearance is 4 per cent, expansion is isentropic, and compression is polytropic with $n = 1.30$.

6.3. Dry air is supplied to the cabin superchargers of an airplane at 14.7 psia and is discharged at 29.4 psia and 275 F. Fifty pounds per minute of this

air is supplied to the primary intercooler of an air-cycle refrigeration system. Determine (a) the air temperature leaving the intercooler if 1425 Btu per min is removed. From the intercooler the air enters a secondary compressor ($n = 1.4$), where the pressure is increased to 46.5 psia. Determine (b) discharge-air temperature and (c) compressor horsepower (mechanical efficiency = 86 per cent). The air then flows through a secondary intercooler, and 1440 Btu per min is removed. Determine (d) the air temperature leaving the secondary intercooler. The air then enters an expansion turbine ($n = 1.3$), where the pressure is dropped to 14.85 psia. Determine (e) the final discharge-air temperature, (f) the work recovered from the expansion turbine (mechanical efficiency = 68 per cent), and (g) the tons of refrigeration developed if air is heated to 75 F.

6.4. A steam-ejector water-vapor system is supplied with motive steam at 150 psia, 98 per cent saturated. Flash-chamber water temperature is 45 F. Make-up water is supplied at 70 F, and the condenser is operated at 2 in. of mercury absolute. The nozzle efficiency is 90 per cent, entrainment efficiency is 65 per cent, and compression efficiency is 78 per cent. The quality of motive steam and flashed vapor at beginning of compression is 90 per cent. Determine (a) pounds of motive steam per pound of flashed vapor, (b) quality of vapor flashed from flash chamber, (c) refrigerating effect per pound of flashed vapor, (d) pounds of motive steam per (hr) (ton refrigeration), and (e) cubic feet of vapor removed from flash chamber per (hr) (ton refrigeration).

6.5. A steam-ejector water-vapor system is operating under the same conditions as in Problem 6.4 with the exceptions that the quality of motive steam and flashed vapor at the beginning of compression is unknown, and the quality of vapor flashed from the chamber is 97 per cent. Determine (a) pounds of motive steam per pound of flashed vapor, (b) quality of mixture of motive steam and flashed vapor at the beginning of compression, (c) pounds of motive steam per (hr) (ton of refrigeration), and (d) Btu per (hr) (ton of refrigeration) removed from the steam mixture in the condenser.

6.6. A steam-jet refrigeration unit is used to supply cooled drinking water directly. Make-up water is supplied to the system at a rate of 20 gph, and drinking water is removed at a rate of 15 gph. If the concentration of salts in the supply water is 18 parts per thousand, determine the equilibrium concentration in the evaporator water.

6.7. A refrigerating machine is to be designed to produce 300 gpm of chilled water at 40 F with supply water at 60 F. Either a steam-jet or a water-vapor centrifugal system may be used. Water vapor entering the compressor or ejector from the flash chamber has a quality of 0.98. Determine (a) required capacity in tons of refrigeration and (b) pounds and cubic feet of vapor removed per minute from flash chamber.

6.8. A centrifugal refrigerating system operates with an evaporator temperature of 40 F and a condenser temperature of 85 F. If it is assumed that the compressor has radial blades and that the theoretical maximum head may be developed, determine the ratio between the required rotative speeds for Freon-11 and water vapor as refrigerants.

6.9. A centrifugal compressor, designed without intercoolers, operates with Freon-11 between the temperatures of 20 and 85 F evaporating and condensing, respectively. The compression efficiency is 75 per cent, and the condenser-water temperature rise is 15 F. If the system produces 200 tons of refrigeration,

determine (a) pounds of refrigerant circulated per (ton) (min), (b) cubic feet of evaporator vapor compressed per (ton)(min), (c) cubic feet of vapor per (ton) (min) discharged from the compressor, (d) theoretical horsepower per ton of refrigeration, (e) coefficient of performance, and (f) gallons per minute of condenser water.

6.10. A heat pump is to be used for heating a building with a design load of 50,000 Btu per hour. Water at 50 F is available as a heat source, and the air supplied to the rooms is to be at 100 F. A 10 F differential is necessary between condensing refrigerant and supply air and between evaporating refrigerant and heat source. If the actual c.p. attained is 70 per cent of that for a reversed Carnot cycle operating between the same temperatures, determine (a) the actual c.p. and (b) the compressor horsepower required to supply this heat load.

6.11. Determine the cost per 1000 Btu of supplying heat in your territory for (a) oil, (b) gas, (c) direct electric heating, and (d) heat pump. In calculating gas and oil costs assume heating-plant efficiencies of 80 and 75 per cent, respectively. In calculating heat-pump costs assume a condensing refrigerant temperature of 110 F, an evaporating refrigerant temperature 10 F below local nonthermal water temperatures, and an actual c.p. 70 per cent of that for the reversed Carnot cycle. Assume a compressor mechanical efficiency of 80 per cent, with the compressor located in the air stream being heated.

6.12. A heat pump is to be designed for the compression distillation of sea water to supply small amounts of potable drinking water. The supply of sea water is 1.25 lb per hour and the discharge from the system is 1.0 lb distilled water and 0.25 lb brine per hour. The evaporator and condenser are combined in one unit, with the water evaporated at 212 F and 0 psi compressed adiabatically and discharged to a condenser coil located in the evaporator. The system is started with an electric heater, after which the heat of vaporization is supplied by the condenser. To increase the efficiency a system of heat exchangers is incorporated with sea water entering at 60 F, drinking water discharged at 80 F, and brine discharged at 70 F. The specific heat of sea water is 0.97 and of brine at discharge is 0.84. Assume steam tables to be correct regardless of dissolved solids and assume no external heat losses. Determine (a) pressure required for compressor discharge, (b) compressor input in horsepower and watts, and (c) electrical input required for equivalent distillation rate accomplished by electrical resistance heating only.

CHAPTER 7

Compression Refrigeration Machines

7.1. Introduction. Compression of a refrigerant gas may be accomplished by any one of several different means. Positive displacement may be used, as in the *reciprocating*, *rotary*, or *gear* types of compressors; centrifugal force may be applied, as in the *centrifugal* compressor; an *ejector* may be used, as in the steam-jet refrigeration cycle; or *absorption* of a low-pressure refrigerant gas in a secondary fluid, followed by the absorbent's release upon application of heat, may be utilized. The basic theory involved in the first three of these methods was discussed in Chapters 4 and 6, and the theory and application of the absorption refrigeration system is presented in Chapter 15. The equipment used in the practical application of positive displacement (reciprocating and rotary compressors) centrifugal, and ejector refrigeration systems is discussed in the present chapter.

7.2. Reciprocating Compressors. The equipment used for compression of the refrigerant forms the heart of the refrigeration unit, and the *positive-displacement reciprocating compressor* is the compression system most commonly used today. Although present-day compressors are fundamentally similar to those constructed in the latter part of the nineteenth century, many refinements have been made. The original reciprocating compressors were slow-speed (50 to 55 rpm) steam-driven devices, and frequently using liquid refrigerant, or in some cases oil injected during the suction stroke, for cylinder cooling. The original units were manually controlled, and small fractional-tonnage machines were unknown. Modern reciprocating compressors are usually electric-motor driven, high-speed (up to 3500 rpm), and usually air- or water-cooled. Low-tonnage and fractional-tonnage machines are usually completely automatic, and large compressors are usually semiautomatic. The modern development of high-speed and fractional-tonnage compressors, together with automatic control, has made possible the extension of refrigeration equipment to a great many applications previously considered impossible. The details of modern reciprocating-compressor design are discussed in the following sections.

7.3. Reciprocating-Compressor Cylinder Design. All reciprocating compressors consist of one or more cylinders suitably valved for suction and discharge of the refrigerant gas and containing reciprocating pistons for compression. The design of the cylinders may vary as to *number*, *arrangement*, and *action* (they may be single-acting or double-acting).

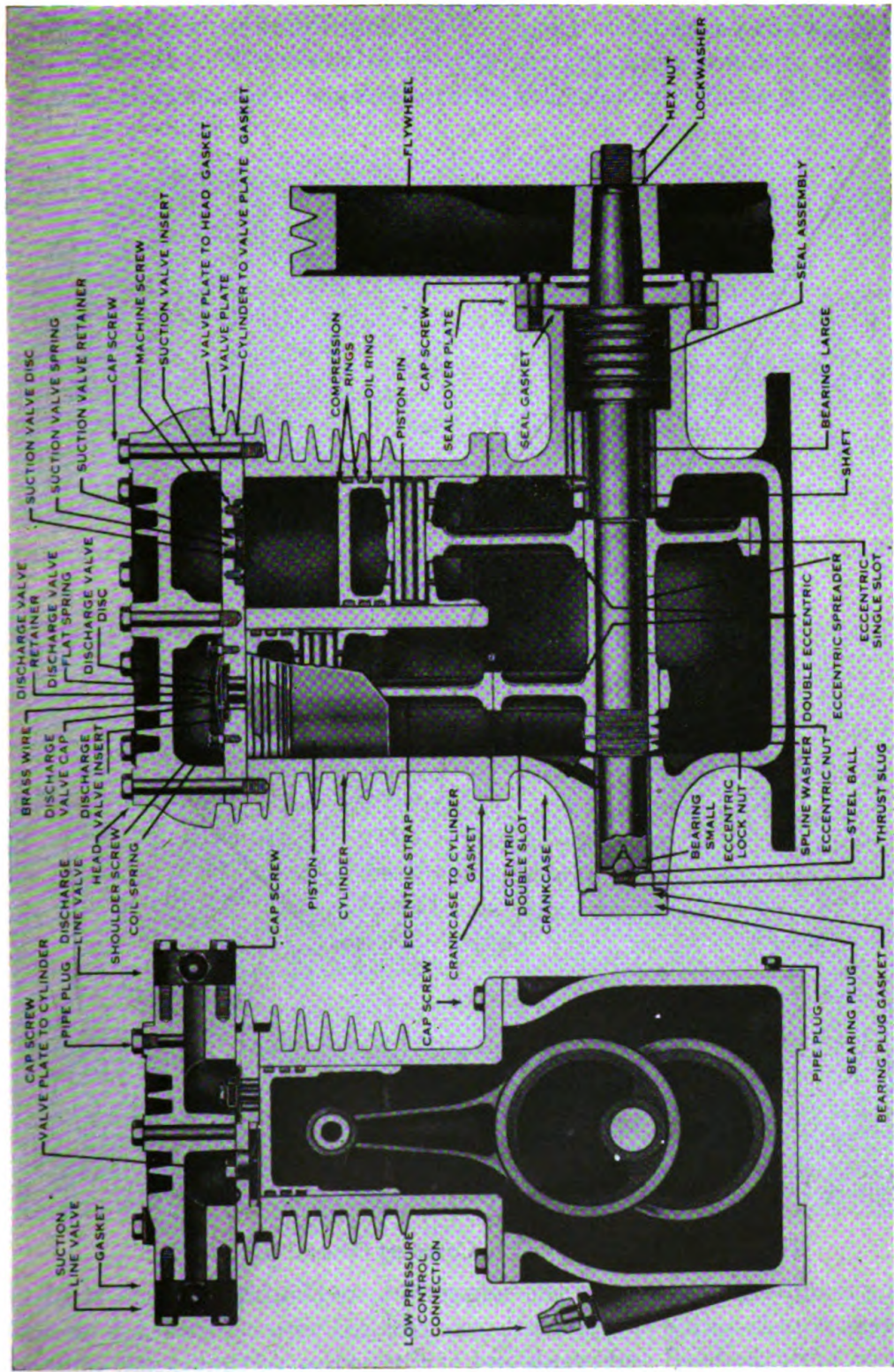


Fig. 7.1. Vertical single-acting compressor. Courtesy Brunner Manufacturing Co.

Cylinder Arrangement. *Single-acting* compressors usually have their cylinders arranged *vertically*, *radially*, or in a *V* or *W*. *Double-acting* compressors usually have their cylinders arranged *horizontally*. *Single-*

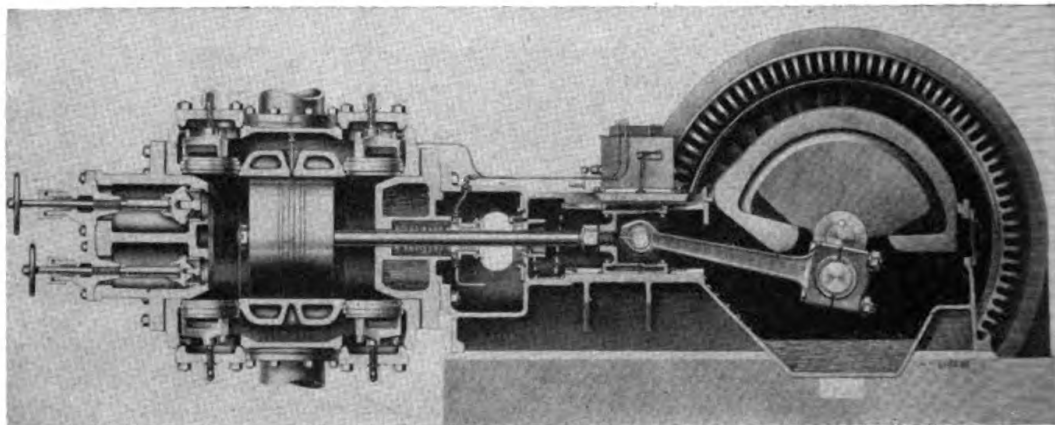


Fig. 7.2. Horizontal double-acting compressor. Courtesy Worthington Pump and Machinery Corp.

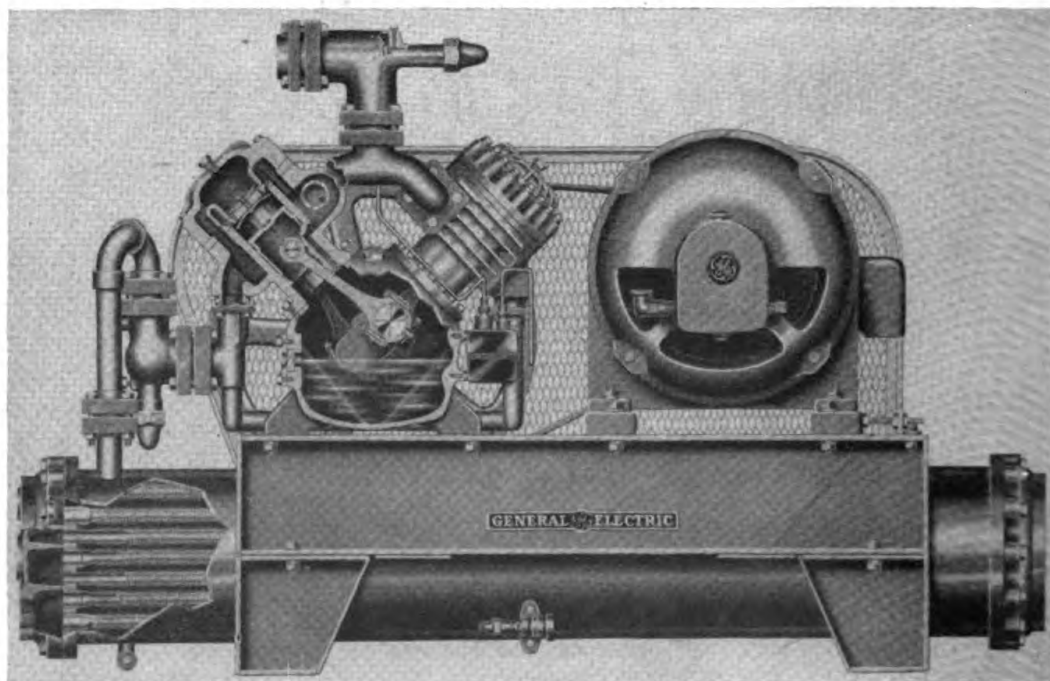


Fig. 7.3. Condensing unit. Courtesy General Electric Co.

acting compressors compress the refrigerant only on the upward stroke, whereas double-acting compressors are arranged with compression chambers on both the head and crank end of the piston, so that a new charge of gas is always being drawn into the compressor while a second charge of gas is being compressed on the opposite side of the piston. Figure 7.1 shows a typical *vertical single-acting* compressor and Fig. 7.2 a *horizontal double-acting* compressor. A *V* cylinder arrangement of a *single-acting*

compressor is shown in Fig. 7.3, and a *radial* compressor is shown in Fig. 7.4. The development of the double-acting compressor has paralleled to a great extent the development of the steam engine, and its design includes stuffing boxes and crossheads similar to those on steam engines.

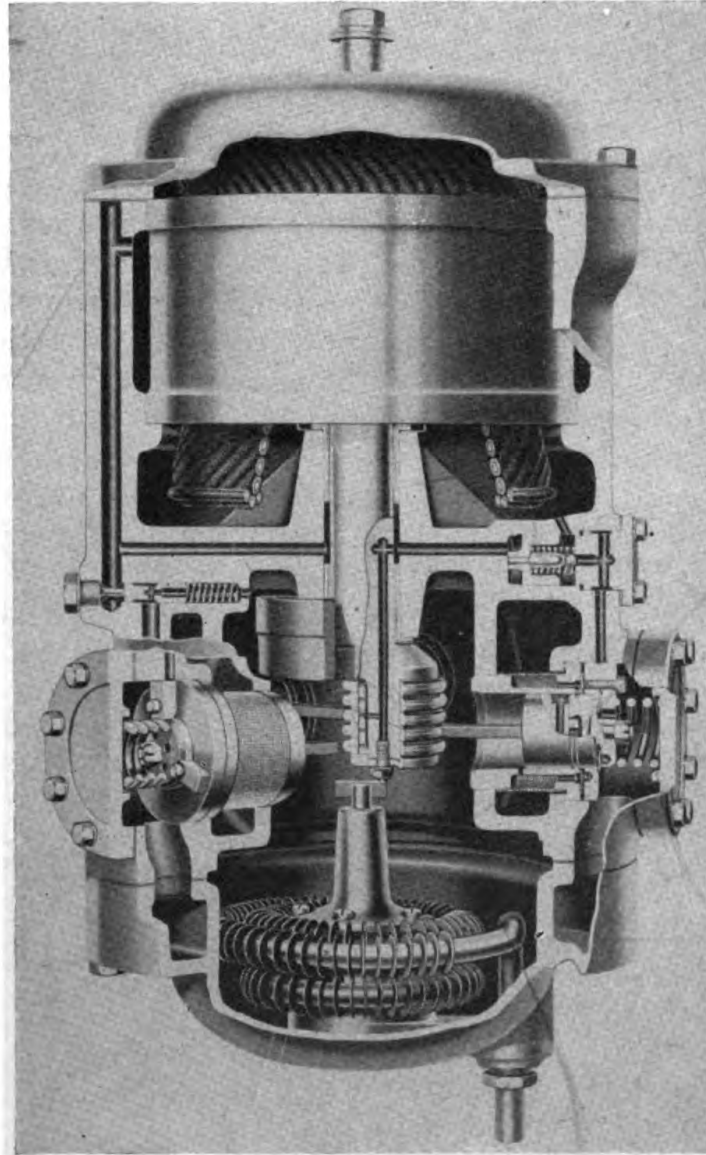


Fig. 7.4. Radial compressor. Courtesy Chrysler Corp., Airtemp Division.

In recent years the horizontal double-acting compressor has almost disappeared from new applications; the single-acting compressor, with its smaller space requirements, higher speeds and greater flexibility, dominates the market.

Number of Cylinders. The design of reciprocating compressors is adaptable to a wide range in the *number* of cylinders. Reciprocating units are available with from one to 16 cylinders with the V, W, and radial

compressors best adapted to the greatest numbers. The present trend toward higher operating speeds is usually combined with low displacement per cylinder, together with the use of several cylinders. Figure 7.5 shows a *single-cylinder vertical* compressor and Fig. 7.6 an arrangement of six pistons for a *W* compressor design.

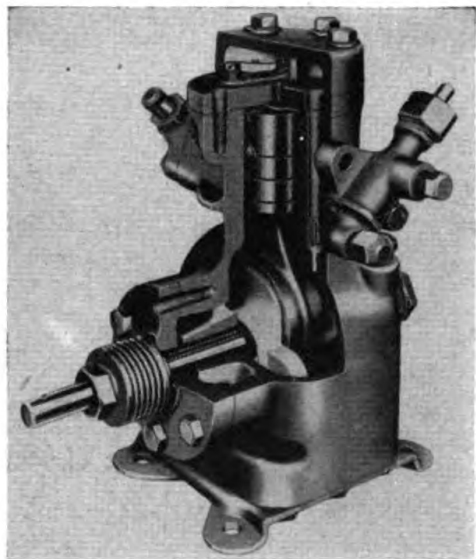


Fig. 7.5. Single-cylinder vertical compressor. Courtesy Universal-Cooler Division, International Detrola Corp.

Cylinder Heads. The *cylinder heads* on most modern compressors are bolted tight to the cylinders, but in some large compressors where there is danger of operation with wet compression or of the entrance of foreign materials into the cylinder space, *safety heads* are provided to reduce the danger of damage. Such a safety head is a secondary head seated at the end of the cylinder and held in position with heavy spiral

springs. If the pressure required to move this head is greater than the

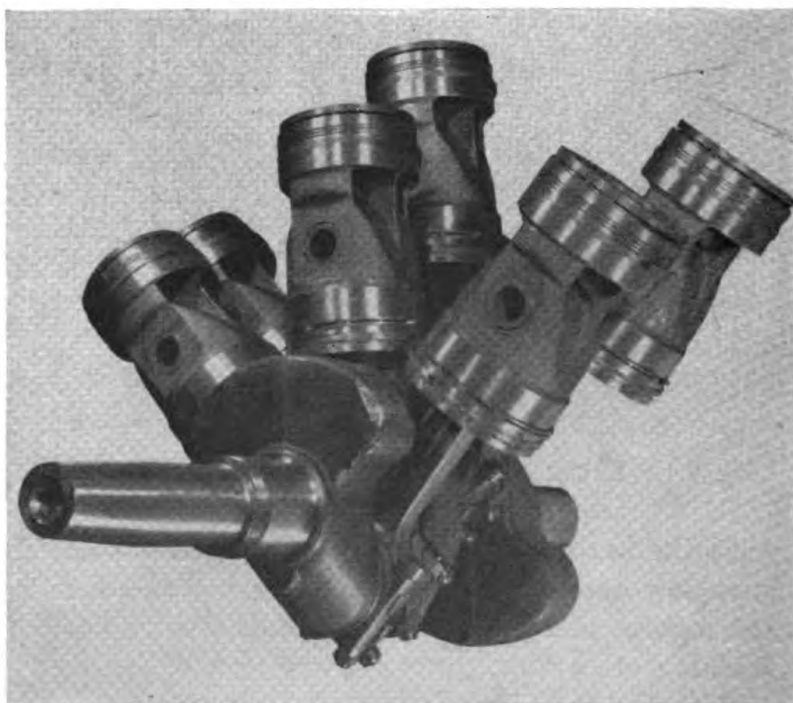


Fig. 7.6. Piston arrangement for *W* design. Courtesy York Corp., York, Penna.

discharge operating pressure, the safety head remains stationary. However, if an excess of wet refrigerant or if foreign materials of any nature enter the cylinder space, movement of the safety head relieves any excessive pressures and thereby prevents serious damage to the cylinder. The extremely small clearances required for high volumetric efficiency in

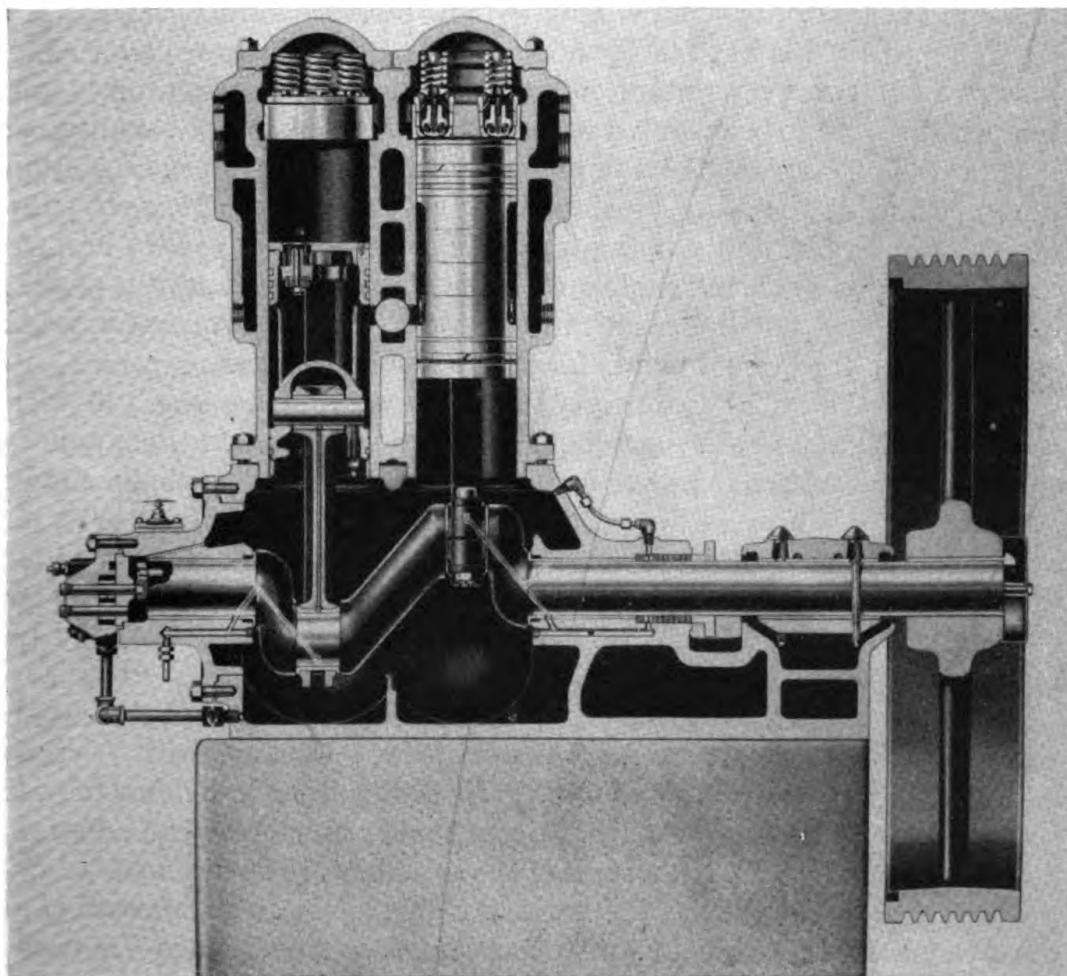


Fig. 7.7. Two-cylinder vertical compressor, showing safety heads. Courtesy York Corp., York, Penna.

the operation of reciprocating compressors (see Chapter 4) frequently makes it desirable to use safety heads in large machines. Figure 7.7 shows a *two-cylinder vertical ammonia* compressor incorporating safety heads.

Small compressors usually rely upon refrigerant filters for the removal of foreign material. In some small-cylinder, high-speed compressors the discharge valve, located in the head of the cylinder, consumes practically the entire head area and therefore functions as both a discharge valve and a safety head.

Multiple-Effect Compressors. The principle of *dual* or *multiple-effect* compression (see Chapter 13) has been known for many years. Such

systems are designed so that on the downward stroke of the piston a charge of gas from the low-pressure evaporator is first drawn into the compressor. Toward the close of the suction stroke a valve or port is uncovered which permits gas from the high-pressure evaporator to enter the cylinder, thus supplying a part of the charge as well as partially compressing the low-pressure evaporator gas and closing the low-pressure suction valve.

This principle is not readily adaptable to one- or two-cylinder compressors. If two evaporators of known loads and at known operating temperatures are to be operated with a dual or multiple-effect compressor, it is possible to design that compressor with a medium-pressure suction port opening at the correct point of the piston stroke. However, the design is inflexible, and varying loads and temperatures cannot be provided for with a single inflexible unit. Even more important, it is almost impossible to predict the exact ratio between the two evaporator loads and therefore the correct points for the two port openings.

Recently the radial compressor, designed with high speeds and with many cylinders, has been suggested as a practical system for the application of multiple-effect compression. Here a portion of the cylinders may be adapted to multiple-effect compression with the remainder used on single compression. Furthermore, comparatively inexpensive cylinder liners containing the secondary ports may be installed or changed after installation so that the balance between compressor and loads may be made under actual operating conditions.

Stage Compression. Stage compression may be accomplished with either *reciprocating* or *centrifugal* compressors. Usually, with reciprocating systems, separate compressors operating from individual power sources are used for each stage. Some multiple-cylinder compressors, however, are so installed that some of the cylinders are operating as a low-pressure stage, the others operating as a high stage. In such a case care must be taken to see that the work of compression on each stage is approximately the same. The theory of stage compression is discussed in Chapter 13.

7.4. Reciprocating-Compressor Drives. Practically all of the early reciprocating compressors were *steam-driven*, usually by Corliss engines, with a few small domestic refrigerators operated by water power. Such equipment is now obsolete, and practically all modern refrigeration compressors are driven by *electric motors*. A few large steam-driven compressors are still being installed and used where steam forms the most economical source of available energy. Some, for use where electric power is inaccessible or for truck transportation and similar purposes, are designed for operation with *internal-combustion engines*.

Open and Hermetic Compressors. Compressors may be further subdivided as *open-type compressors* and as *hermetic compressors*. With the open type, power is received from an external source with one end of the

crankshaft extending through the crankcase to either a direct motor drive or a V-belt, gear, or chain drive. With such a design it is necessary to seal the crankcase against refrigerant leakage at the point where the shaft emerges. In practically all cases the crankcase is exposed to the circulating refrigerating gas, and leakage around the shaft must be prevented under both static and moving conditions. With horizontal double-acting compressors the piston rod slides back and forth through a stuffing box, usually sealed with asbestos and graphite or a metallic or semimetallic packing. Most compressors designed with a rotating shaft emerging from the crankcase utilize a bellows or sylphon seal of which there are several designs. One such design, shown in Fig. 7.8, consists of a gasket and flexible spring-expanded bellows attached to the crankshaft with a ring on the bellows rubbing against a smooth bearing surface on the crankcase. With a reasonable spring pressure of 30 to 50 lb per square inch and with normal lubrication, refrigerant leakage is reduced to a minimum both under operation and stationary conditions. Applications of this type of seal are shown in Figs. 7.1 and 7.5.

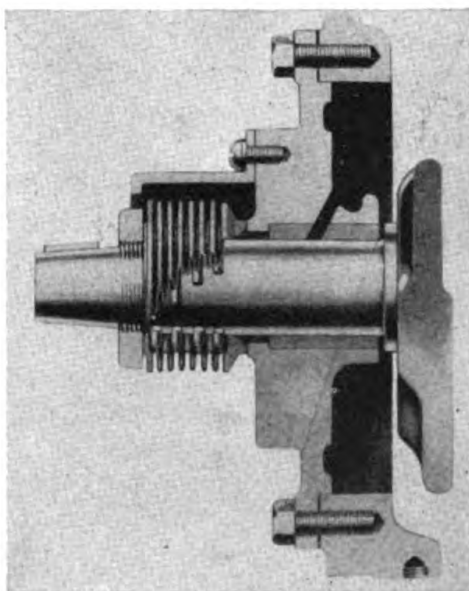


Fig. 7.8. Bellows crankshaft seal.
Courtesy Universal Cooler Division,
International Detrola Corp.

The *hermetic* type of compressor may be either reciprocating or rotary and is a direct-drive unit with both the motor and the compressor hermetically sealed within the housing. This arrangement eliminates the necessity of any shaft seal and consequently prevents refrigerant leakage at that point. It also reduces the operating noise, because the external drive is also eliminated. It has the disadvantage, however, that the moving parts are inaccessible for repairs. Lubrication is greatly simplified, since both the motor and the compressor operate in a sealed space with the lubricating oil. Figure 7.9 shows a typical hermetic reciprocating compressor and Fig. 7.10 a typical hermetic rotary unit.

Compressor Speeds. The trend in modern compressor design is toward higher operating speeds. Whereas the original steam-driven reciprocating compressors operated at approximately 50 rpm, modern compressors operate from 250 to 3400 rpm. Recently some designs of compressors have been operated experimentally at speeds as high as 5000 rpm with satisfactory results. The advantages of high-speed operation are obvious, because each increase in compressor speed reduces the displacement

required per stroke and consequently the compressor size. In the design of high-speed compressors considerable care must be taken to reduce throttling through the suction and discharge valves. An increase in the

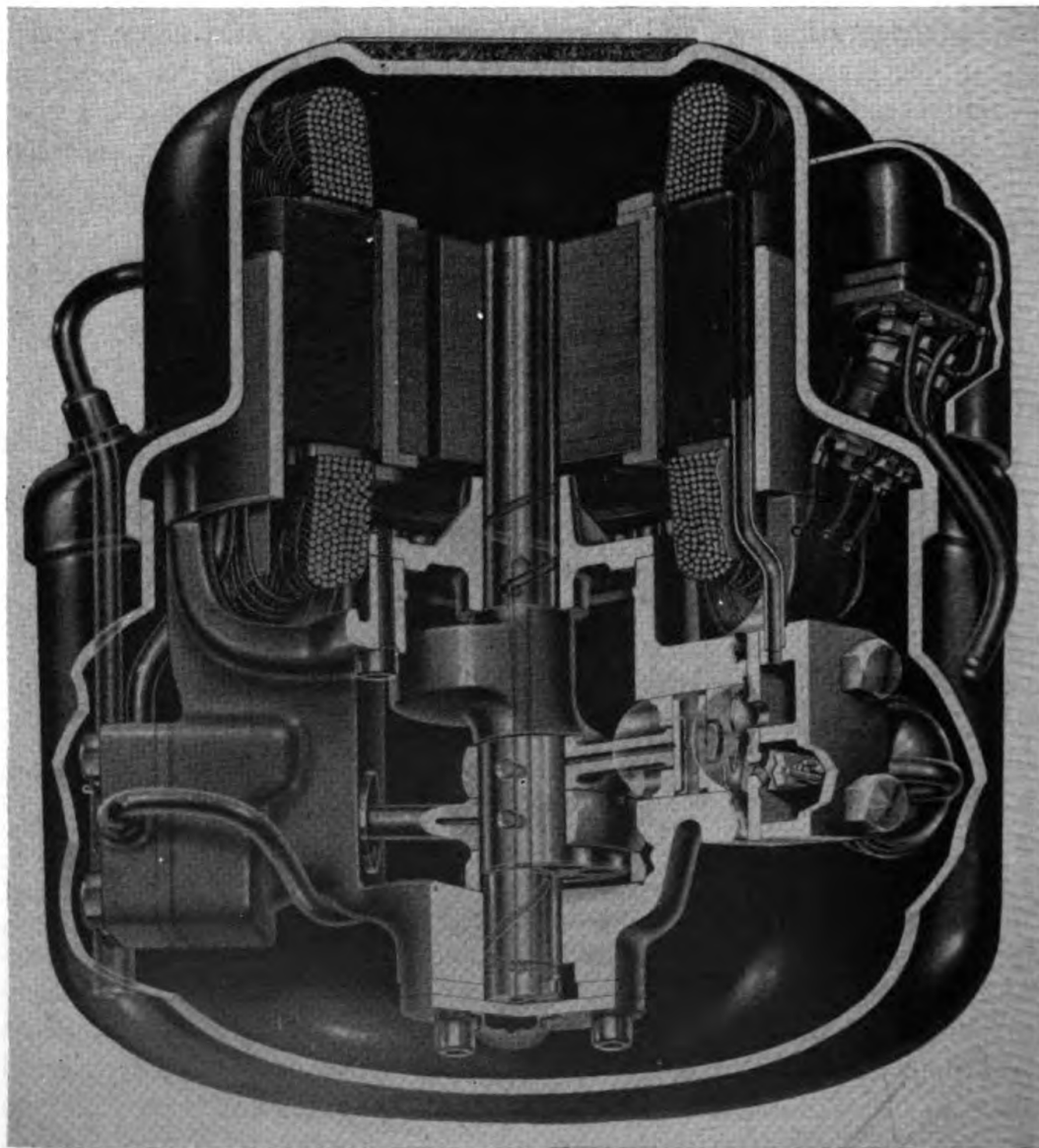


Fig. 7.9. Hermetic reciprocating compressor. Courtesy Universal Cooler Division, International Detrola Corp.

pressure differential required for valve operation will result in an increase in the work of compression as well as a decrease in the clearance volumetric efficiency.

Bearings and Crankshaft. Compressor *bearings* may be either of the *sleeve* type or the *antifriction* type. The latter is usually limited to the main bearings alone. The crankshafts of larger compressors are usually of the *crank-throw* type, made of drop-forged steel and frequently case-hardened. However, if the bearing loads are low, small compressors may

be designed with *eccentrics* in place of crank throws. These eccentrics, usually made of cast iron, are fastened to a straight shaft with the bearing end of the connecting rod completely encircling the eccentric. Figures 7.3 and 7.6 show compressors incorporating the *crank-throw* or *offset* type of crankshaft, and Figs. 7.1 and 7.5 show compressors incorporating the *eccentric* type of crankshaft.

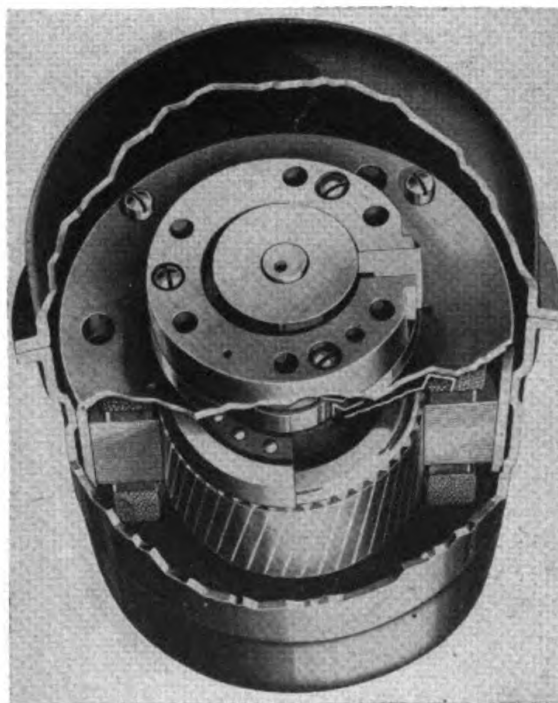


Fig. 7.10. Hermetic rotary compressor.
Courtesy Norge Division, Borg-Warner Corp.

Compressor Capacity Control. Control of compressor capacity may be obtained in several ways. This subject is discussed in Chapter 14.

7.5. Reciprocating-Compressor Valves. All refrigeration compressor *suction* and *discharge valves* are dependent for their operation upon a difference in pressure between the inside of the cylinder and the suction or discharge line. Mechanically operated valves would reduce valve throttling, if they could be applied properly, but they have proved unsatisfactory because each change in the condenser or evaporator operating pressures requires a change in the valve setting. With the pressure-differential types of valves the suction valve opens when the cylinder pressure reaches a point slightly below the suction-line pressure, and the discharge valve opens when the compressor pressure exceeds slightly the discharge-line pressure. Thus the compression range is from somewhat below the evaporator pressure to somewhat above the condenser pressure. The pressure differentials required for operation of the valves depend upon the valve design and the compressor speed.

Valve Arrangement. The suction and discharge valves may be arranged either with both placed in the compressor head or with the suction valve in the top of the piston and the discharge valve in the compressor head. With the latter design the suction gas may be admitted either through the cylinder wall or through the compressor crankcase. The *uniflow* arrangement with the suction valve in the head of the piston and the discharge valve in the head of the cylinder is the usual design for

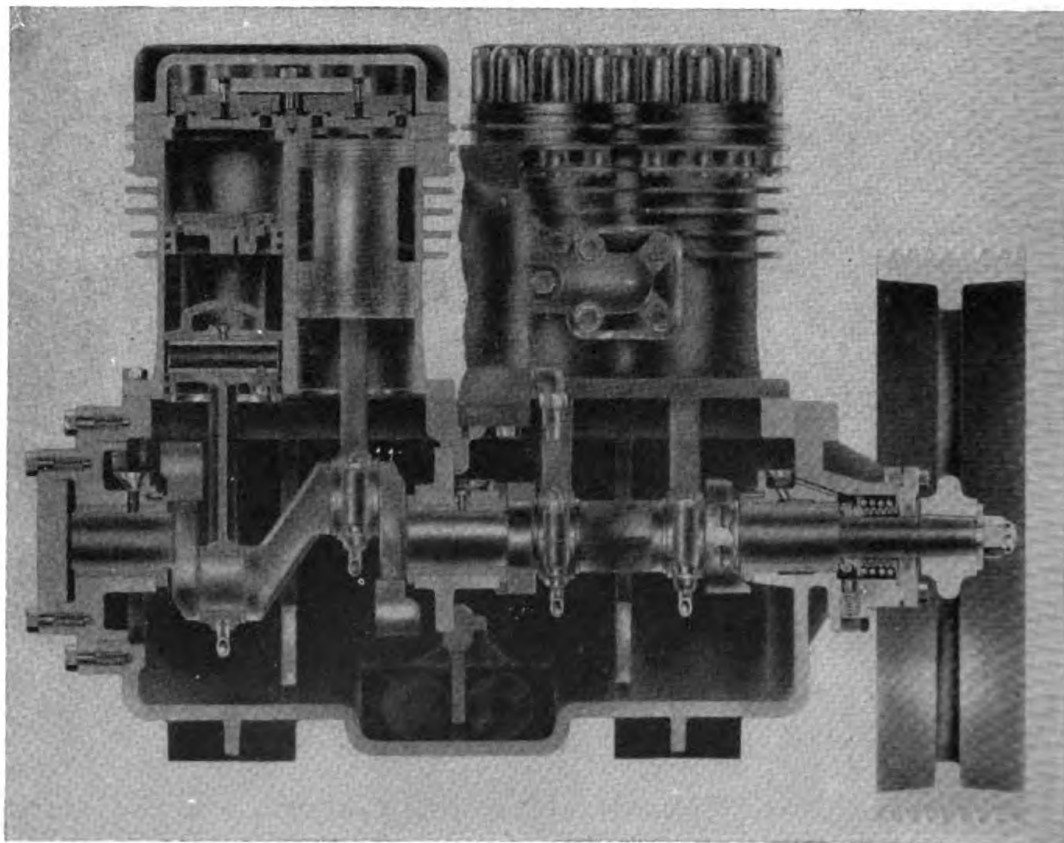


Fig. 7.11. Vertical compressor with uniflow valve arrangement. Courtesy Carrier Corp.

large ammonia compressors. Arrangements with both suction and discharge valves in the cylinder head are usual with small or fractional-tonnage compressors. Figures 7.7 and 7.11 show compressors with *uniflow* valve arrangements and Figs. 7.1 and 7.5 compressors with both suction and discharge valves in the cylinder heads.

Valve Design. Compressor suction and discharge valves may be classed as either *poppet*, *ring-plate*, or *flexing*. All operate upon pressure differential, and there are many modifications of each class. Typical multiple *poppet-type* suction valves for a uniflow arrangement are shown in Fig. 7.12. A typical 6-in., three-ring *ring-plate* valve is shown in Fig. 7.13. Typical *flexing* valves for fractional-tonnage compressors are shown in Fig. 7.14. On small compressors both the suction and dis-

charge valves are usually mounted on valve-plate assemblies that may be removed as a unit after the head of the compressor is unbolted.

7.6. Reciprocating-Compressor Lubrication and Cooling. Lubrication of early low-speed compressors was simple because intermittent

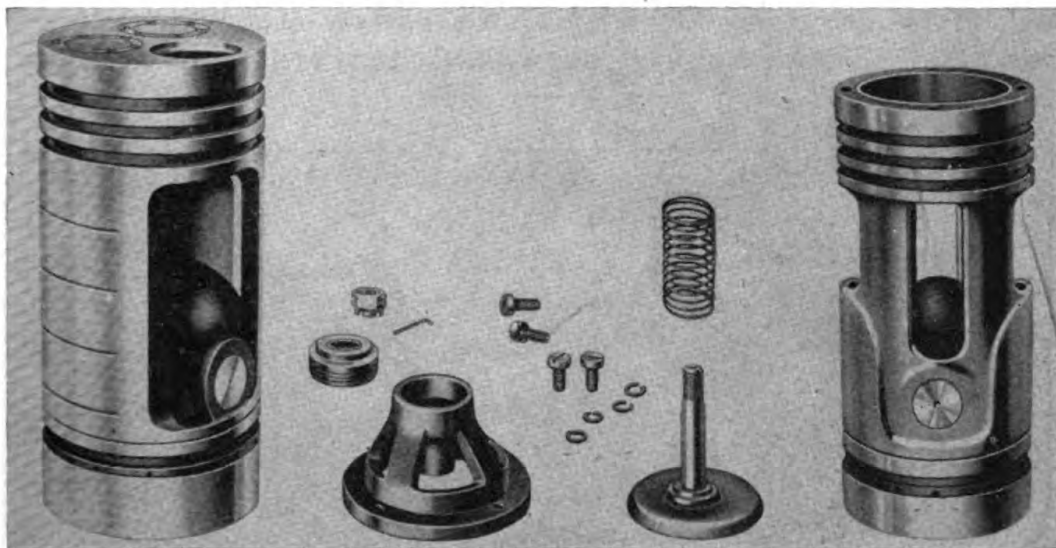


Fig. 7.12. Poppet-type suction valves. Courtesy York Corp., York, Penna.

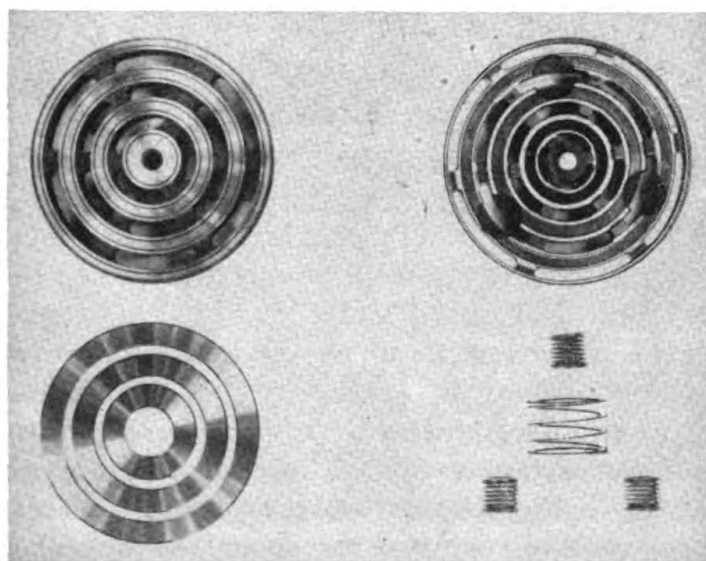


Fig. 7.13. 6-inch, 3-ring, ring-plate valve. Courtesy J. H. H. Voss Co.

oiling by the operator was sufficient. Some compressors were lubricated by the injection of oil on each stroke at the same time that the suction gas was drawn into the compressor. Thus the lubricant served both for lubrication of the moving parts as well as for cooling of the compressor. These methods are of course completely unsatisfactory for modern high-

speed compressors. Two systems, *splash lubrication* and *forced lubrication*, are common at present. In splash lubrication, with each rotation of the shaft the crank and connecting rod dip into the crankcase oil reservoir and thereby splash the lubricant into openings that will allow it to work into the moving parts. Large compressors are usually equipped with forced lubrication systems involving a separate oil pump to maintain circulation. The oil is usually filtered and distributed to the bearings, crankshaft, connecting rods, and wrist pins. A typical small unit using

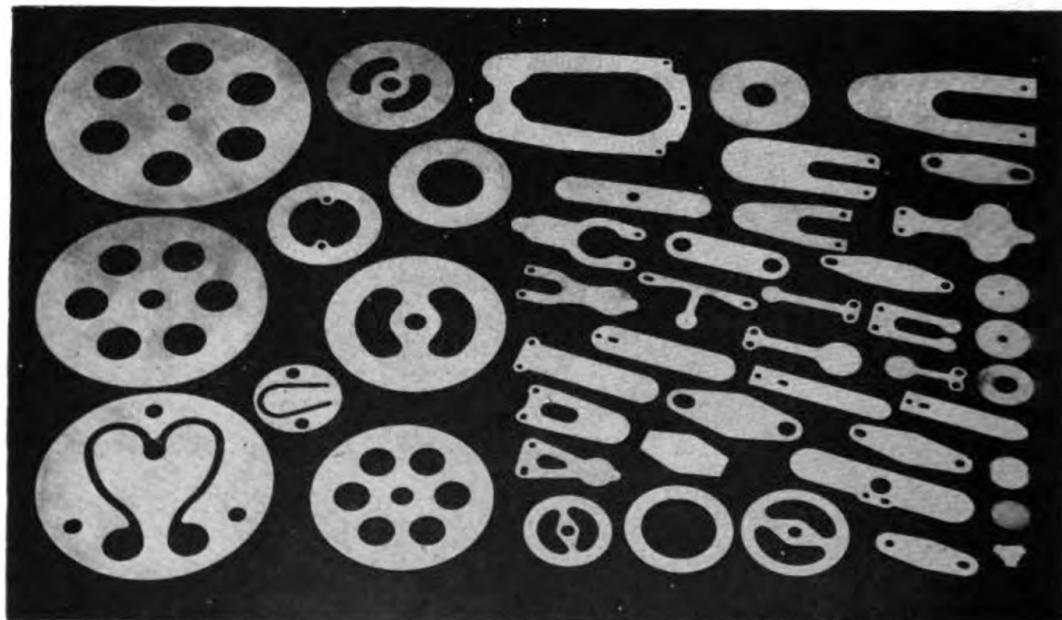


Fig. 7.14. Various designs of fractional-tonnage compressor flexing valves. Courtesy Detroit Stamping Co.

splash lubrication is shown in Fig. 7.3, and modern compressors using *forced lubrication* with rotary-gear oil pumps attached to the end of the crankshaft are shown in Figs. 7.7 and 7.15.

In a refrigeration system in which the lubricant comes into direct contact with the refrigerant, it is necessary that the lubricant be as *free from moisture* as possible, possess a very low *acidity*, not exhibit the phenomenon of *wax separation* upon a lowering of the temperature, and possess a relatively high *flash point* and a *low sulfur* content. It should also have a *pour-point* temperature low enough to prevent congealing if the oil reaches the low-temperature portions of the system and yet be able to lubricate satisfactorily at high temperatures. Thus a *highly refined* and properly treated oil must be used. When the refrigerant comes in contact with the lubricant, refrigeration-compressor operation involves some additional problems not normally found in the lubrication of other equipment.

The *degree of solubility* of the refrigerant in the oil varies with the refrigerant. For example, with sulfur dioxide and the Freon refrigerants

the solubility is low, whereas with methyl chloride it is quite high (see Chapter 5). This solubility gives rise to the phenomenon of *oil foaming*, a term used to describe the release of dissolved refrigerant from the lubricating oil upon a sudden drop in crankcase pressure or a sudden rise in temperature. When oil foaming occurs, extensive loss of oil from the

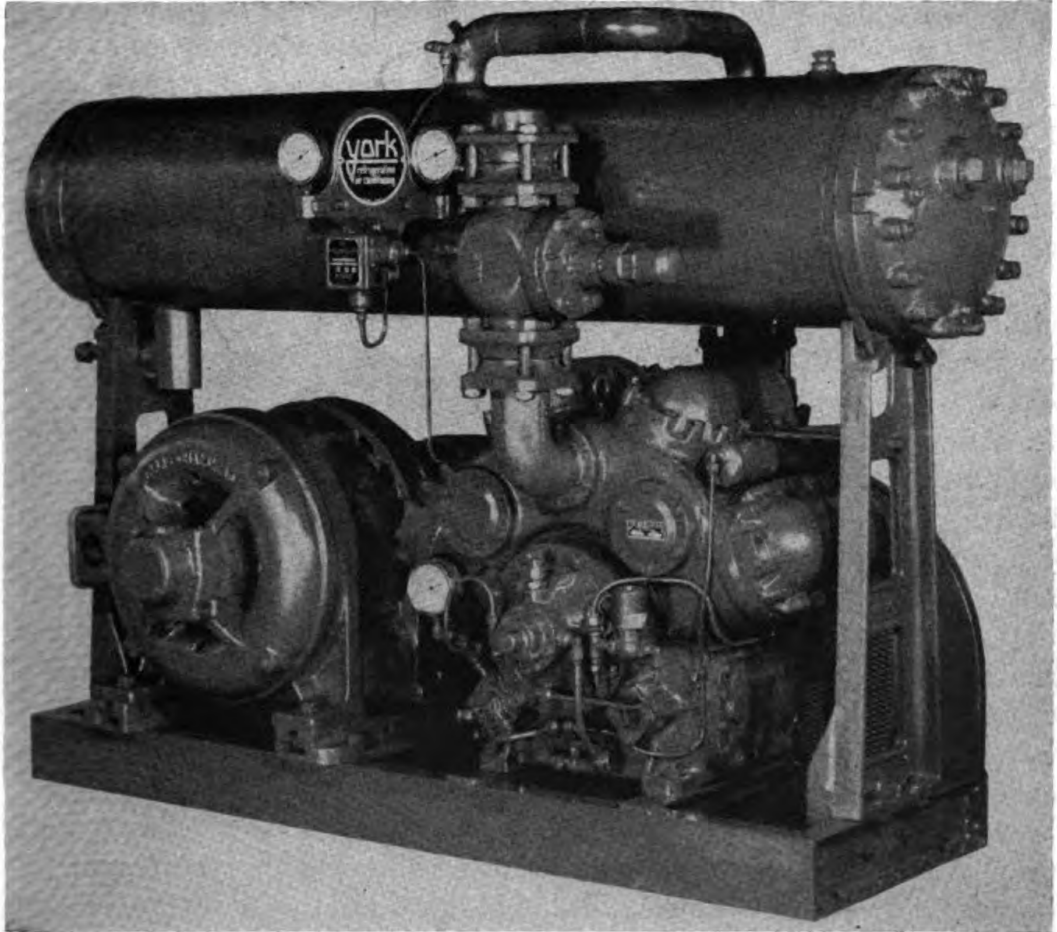


Fig. 7.15. W design water-cooled compressor with forced lubrication. Courtesy York Corp., York, Penna.

crankcase may take place, with considerable oil being carried with the refrigerant into the condenser and eventually into the evaporator. If the loss of oil from the crankcase becomes excessive, unsatisfactory lubrication may result, as well as oil locking of the evaporator and possibly also of the return lines to the compressor. In addition, the excess of oil in the evaporator or the condenser decreases the coefficient of heat transfer. Even though oil foaming is not encountered, a certain amount of lubricant usually reaches the condenser and evaporator and is eventually returned by normal means to the compressor. However, if it is particularly important that no oil reach the evaporator, an oil separator may be installed between the discharge of the compressor and

the condenser to collect and return the lubricant to the crankcase (see Chapter 14). It is evident that the lubricant may be subjected to both the low temperatures of the evaporator as well as the high temperatures at discharge from the compressor, and that for satisfactory operation the lubricant must operate satisfactorily over this wide range of temperatures.

7.7. Reciprocating-Compressor Cooling. Many early compressors were cooled by operating with a *wet compression* cycle (see Chapter 4). Despite the slightly higher coefficient of performance of the wet as compared with the dry compression cycle, the reduced capacity of the compressor, together with the danger of an excess of liquid refrigerant in the clearance spaces, causing compressor damage unless safety heads are used, has practically eliminated this means of cooling in modern compressors.

Large compressors are usually *water-cooled*, with the water jacket either cooling the cylinder walls or both the cylinder walls and the compressor head. Modern small compressors are either *water-cooled* or *air-cooled*, with extended finned surfaces cast on the exterior of the cylinder. In a few cases small compressors may be found in which there is no attempt at any purposive cooling other than through transfer from non-finned surfaces to the lower temperature ambient air. A water-cooled compressor usually runs lower in temperature than an air-cooled unit, but even under the best conditions cylinder cooling removes only a portion of the superheat of the refrigerant gas. However, whatever superheat is removed from the gas results in some decrease in the work of compression as well as a reduction in the load on the condenser. Typical water-cooled compressors are shown in Figs. 7.7 and 7.15, and typical air-cooled compressors in Figs. 7.3 and 7.11.

7.8. Rotary Compressors. In recent years *rotary* compressors, usually *hermetically sealed*, have become quite popular for fractional-tonnage refrigeration applications. In general, the design of rotary compressors may be divided into two classifications: (1) those in which the sealing blades rotate with the shaft and (2) those using one or more stationary blades for sealing the suction from the discharge gases. The *single-blade* rotary compressor (see Fig. 7.10) is the more popular. Here an eccentric driven rotor revolves within a housing in which the suction and discharge passages are separated by means of a sealing blade. When the rotating eccentric first passes this blade and the suction opening, the compressor suction space is very small. As the eccentric rotates, this crescent-shaped space becomes increasingly larger and thereby draws in a charge of suction gas. When the eccentric again passes the blade, the gas charge is cut off from the suction inlet, compressed, and discharged from the compressor. Thus the action of compression and suction is continuing at the same time, with the compression and the suction spaces separated by the sealing blade.

There are many modifications of detail in the design of rotary compressors, but practically all operate upon the same general principle. Rotary compressors have the advantage of being quiet in operation and reasonably free from vibration. Refrigerants requiring a reasonably high volumetric displacement per ton of refrigeration may be used satisfac-

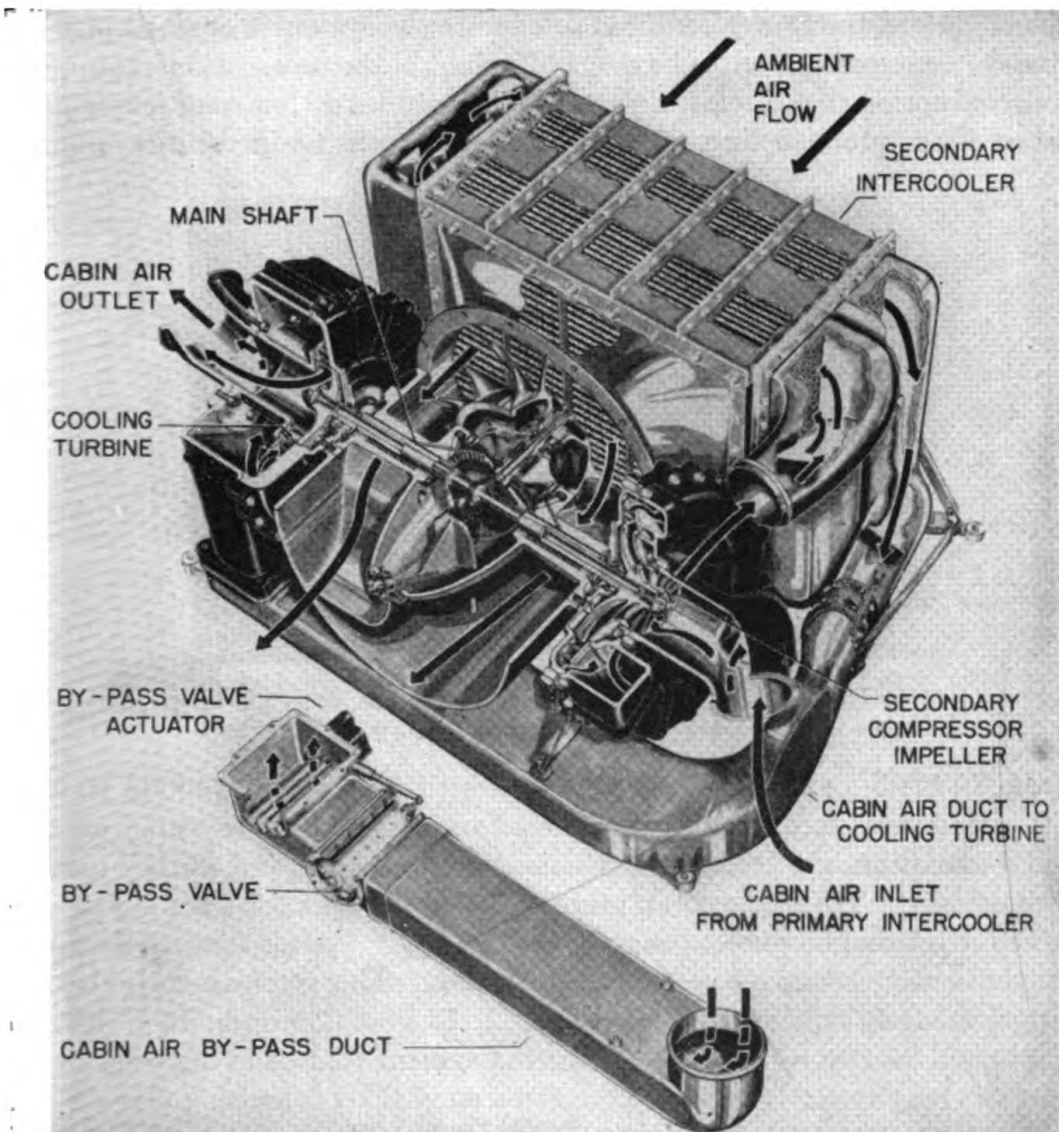


Fig. 7.16. Air-cycle refrigeration unit for Lockheed Constellation. Courtesy Lockheed Aircraft Corp.

torily with this type of unit. Rotary compressors do not deliver as great a volumetric displacement as centrifugal compressors of comparable size, but the displacement is usually higher than that of comparable reciprocating compressors.

7.9. Air-Cycle Refrigeration Equipment. *Air-cycle* refrigeration equipment, as originally developed in the nineteenth century and as used

to some extent up until the 1920's, is now obsolete. Although stationary air compressors and expanders for isentropic air expansion and cooling form a workable means of obtaining refrigeration, such a system becomes economically impractical in competition with other forms of refrigeration (see Chapter 6). During and since World War II, however, application has been made of air-cycle refrigeration to aircraft air-conditioning systems. Here the power required to operate the system is of small importance in comparison with the size and weight of the unit. It is a common misconception that aircraft flown at high altitudes do not require refrigeration for comfort cooling. In the summer months temperatures ranging

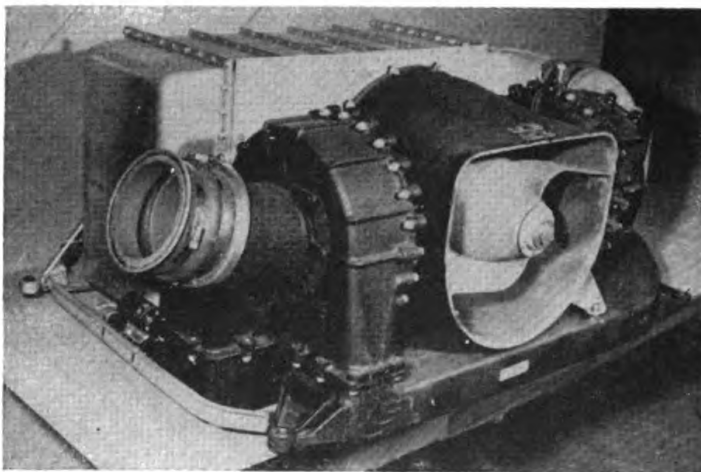


Fig. 7.17. Lockheed Constellation air-cycle unit. Courtesy Lockheed Aircraft Corp.

from 70 to 85 F are often encountered at altitudes of from 7000 to 10,000 ft. Moreover, large quantities of heat are supplied to the cabin air by solar radiation, electrical and mechanical equipment, and the occupants themselves. There is also an increase in temperature of the moving air when it is brought to rest (relative to the airplane) after passing through a ventilation scoop or a fuselage opening. This temperature rise is approximately equal to 6 F at 180 mph, 11 F at 250 mph, 16 F at 300 mph, and 29 F at 400 mph.¹ Thus it is evident that even unpressurized aircraft require refrigeration under some conditions, whereas pressurized-cabin aircraft operating at high altitudes, where the work of compression results in an additional air-temperature increase, require even more refrigeration and for greater periods of time. Ground refrigeration is required frequently, since considerable time elapses between loading at the air terminal and the point of take-off, during which the planes are exposed to ground ambient temperatures. This period may vary from 5 min for a two-engine transport to as much as 10 or 15 min for a four-

¹ Fisher, John T., "High Speed Airliners Need Refrigeration." *Refrigerating Engineering*, Vol. 52, No. 4 (October, 1946), p. 303.

engine transport, and unavoidable ground delays may increase it to as much as 30 min.

Several different refrigeration systems have been applied to aircraft with varying degrees of success. A *vapor-compression* refrigeration system proposed for the Lockheed Constellation² would weigh approximately 58.5 lb per ton of refrigeration. This airplane would require a

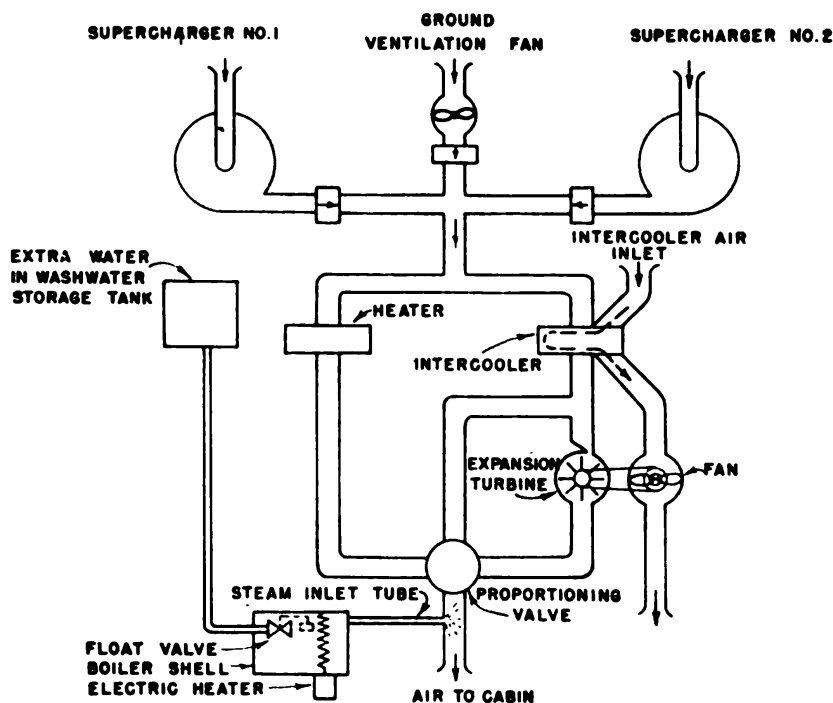


Fig. 7.18. Cabin air-conditioning system for DC-6 Transport.
Courtesy Harrington Manufacturing Co.

10-ton vapor-compression unit with a 17-hp compressor motor and a 5-hp motor for the condenser fan. An additional disadvantage would be the poor accessibility and removability of the refrigeration system. A *dry ice* (solid-phase carbon dioxide) refrigeration system for the same plane would weigh approximately 131 lb per ton of refrigeration at the start of a cross-country flight and, after half the flight time had elapsed, approximately 88 lb per ton. Maintenance difficulties and poor removability would again be undesirable features, and the additional problem of handling dry ice would be involved.

When compared with these two refrigeration systems an *air-cycle* unit has several inherent advantages. In a pressurized plane, cabin superchargers are already required equipment and are needed primarily for high-altitude flying when refrigeration is least necessary. Thus at intermediate altitudes a portion of the supercharger capacity is available

² Messinger, Bernard L., "Refrigeration for Air Conditioning Aircraft." *Refrigerating Engineering*, Vol. 51, No. 1 (January, 1946), p. 21.

for refrigeration purposes, and the superchargers need not be any larger in capacity than when used for pressurization alone. The air-cycle refrigeration system designed for the Lockheed Constellation and actually used in it has a capacity of approximately 10 tons, and the weight of the unit with all necessary accessories chargeable to the refrigeration system is approximately 250 lb. Therefore the weight of the system per ton of refrigeration is 25 lb. In addition to low weight, an air-cycle refrigeration system has the advantage that the equipment can be con-

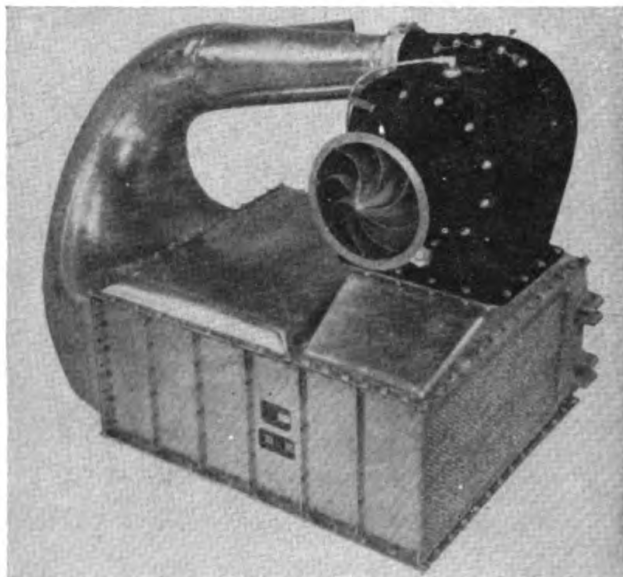


Fig. 7.19 DC-6 Air-cycle refrigeration unit.
Courtesy Harrington Manufacturing Co.

tained in a single removable unit with only air inlet and air outlet connections. No mechanical or electrical power is required. It can be located in an easily accessible part of the plane, refrigerant leakage is unimportant, and the unit is easily repairable.

7.10. Aircraft Air-Cycle Refrigeration Equipment. Figure 7.16 shows a cutaway sketch of the air-cycle refrigeration unit used in the Lockheed Constellation, and Fig. 7.17 is a photograph of the same system. The air from the cabin *superchargers* is introduced into the *secondary compressor* (Fig. 7.16) for additional compression, with an accompanying rise in air temperature. From here the air passes through the *secondary intercooler*, where a portion of the heat is removed by forced convection. The air is then introduced into the *expansion-cooling turbine*, where the air is expanded isentropically and the work of expansion is recovered to supply the necessary power for the secondary compressor and the intercooler fans. No additional shaft work is required for operation of these units, since only partial compression occurs in the secondary compressor. The entire unit weighs 150 lb and is remov-

able from the airplane whenever refrigeration is not required. A cabin-air by-pass duct and by-pass valve, also shown in Fig. 7.16, is so arranged that any proportion of the air may be automatically by-passed, dependent upon the cabin cooling demands.

Figure 7.18 shows schematically the cabin air-conditioning system used in the American Airlines DC-6 transports. This unit includes a

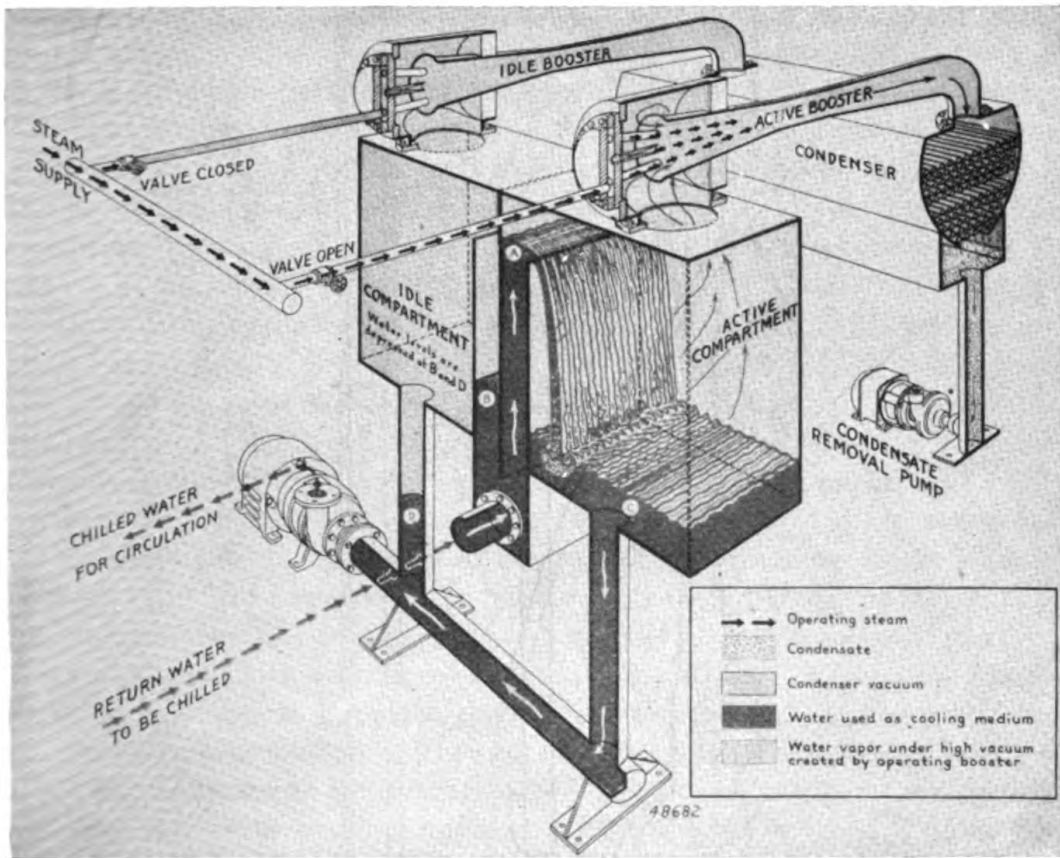


Fig. 7.20 Steam-jet water-cooling system. Courtesy Ingersoll-Rand Co.

steam humidifier to provide sufficient moisture to maintain the desired humidity level within the airplane. Figure 7.19 shows the air-cycle refrigeration *expansion turbine*, fan, and *aftercooler* designed for this airplane. Here again the work of the expansion turbine is used to drive an axial-flow fan used to draw air through the aftercooler.

7.11. Steam-Ejector Refrigeration Equipment. *Steam-jet* refrigeration systems, the theory of which was described in Chapter 6, have their primary application in high-temperature water cooling and air conditioning. They are particularly adapted to air-conditioning systems because of the complete safety of water as a refrigerant, their freedom from vibration, and their ability to adjust quickly to load variations. The major disadvantage involves the maintenance of the high vacuums necessary

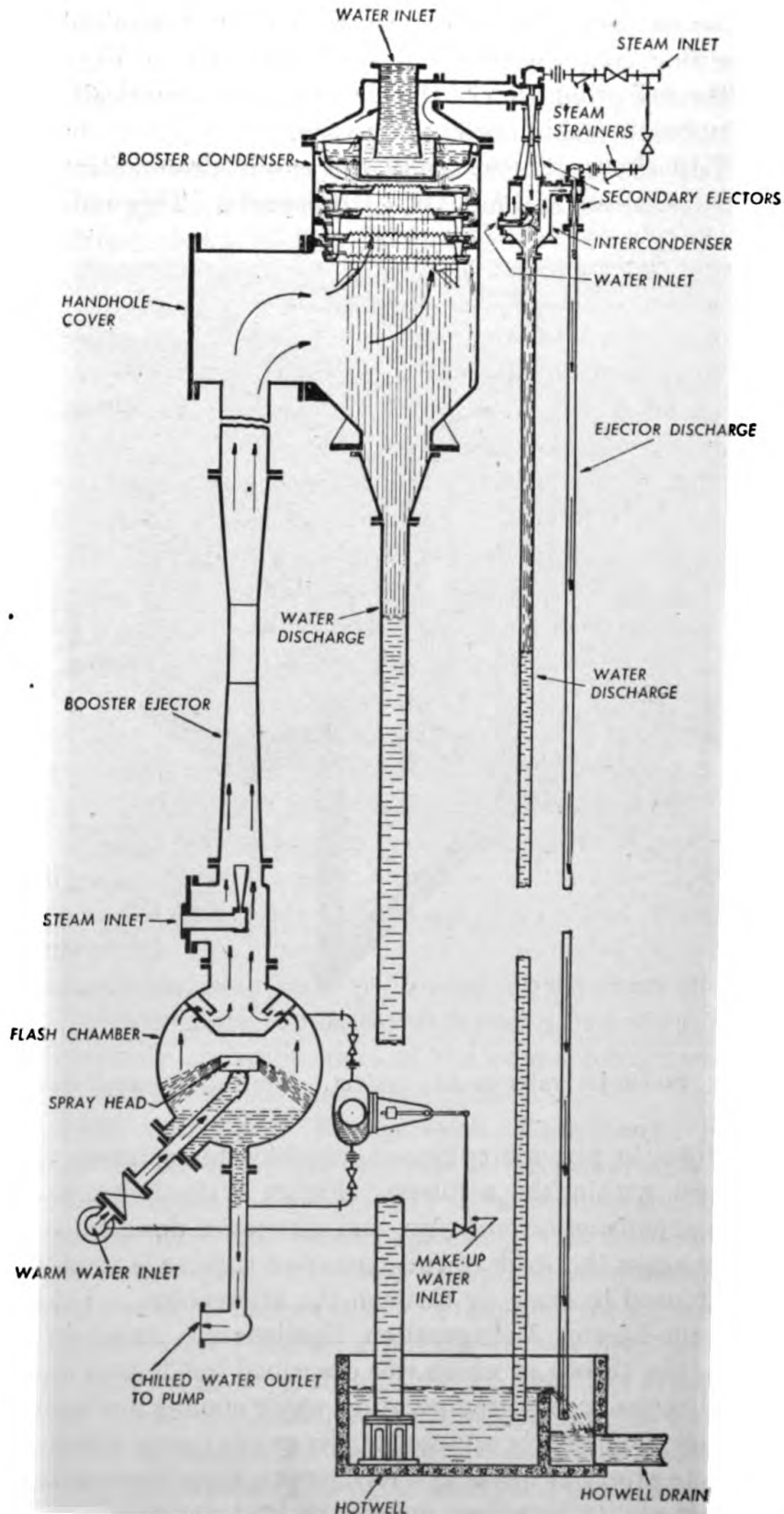


Fig. 7.21. Steam-jet unit with barometric condenser. Courtesy Elliot Co.

for proper operation. Controls, which were unsatisfactory on early equipment, have been improved, and by 1947 there were more than 22 installations, ranging up to 300 tons capacity, in New York City alone. Whether steam-jet refrigeration can be applied to economic advantage depends greatly upon the availability of a high-pressure steam source and a condensing-water source in large volume and at reasonably low rates. In some instances steam-generating equipment may be available and unused during the summer months when cooling is desired, or advantageous off-season district-heating steam rates may be available. Both of these factors help to determine the economic feasibility of such a system.

The performance of steam-jet equipment, when measured by the pounds of motive steam required per (hour)(ton of refrigeration), improves rapidly with increasing pressure of the available steam. For example, in a typical system with a chilled-water temperature of 50 F, approximately 44 lb of motive steam is required per (hour)(ton of refrigeration) if the available steam is at 25 psia, approximately 27 lb of motive steam at 50 psia, 21 lb at 75 psia, 18 lb at 100 psia, and 16 lb at 150 psia. All these figures increase with a lowering of the evaporator chilled-water temperature. There is little gain to be realized by increasing the motive steam pressure above 100 to 150 psia.

Figures 7.20 and 7.21 show diagrammatically two typical steam-jet water-cooling systems. In addition to the high-pressure steam source, consisting of the *evaporator*, the *booster*, and the *primary condenser*, it is necessary to provide *secondary ejectors* and *condensers* in order to produce the initial evacuation of the system during operation. *Pumping equipment* is necessary to circulate the chilled water, which serves as both a primary and a secondary refrigerant, to the point of the cooling load.

7.12. Centrifugal Refrigeration. A *centrifugal* refrigeration system operates upon the vapor-compression refrigeration cycle. Unlike the reciprocating and rotary machines, it is dependent upon centrifugal force and not positive displacement for compression of the gas. The theory of its operation was discussed in §6.5. Centrifugal units are particularly well adapted to large-capacity systems (up to 3000 tons), although it is also possible to secure units as low as 50 tons in rating. Because centrifugal units operate best with refrigerants possessing a high specific volume, they are frequently used for extremely low-temperature applications. They are adaptable to a wide range of temperatures from -130 F to 50 F. One important advantage of centrifugal machines is their flexibility under varying loads. Units may be designed to operate at capacities as low as 20 per cent of normal load with reasonable efficiency.

Centrifugal refrigeration compressors are particularly well suited to *direct steam-turbine* drive because of their high operating speeds. The water-cooling equipment of one manufacturer is designed to be operated between 3500 and 4000 rpm for units developing 1000 to 2000 tons capac-

ity and from 7000 to 8000 rpm for units developing 100 to 200 tons capacity. However, a great many applications, particularly in the smaller sizes, are driven by electric motors. Power may be provided by motors of the *squirrel-cage*, *slip-ring*, or *synchronous* type designed to operate at normal speeds but equipped with standard gear-type speed increasers. In a few applications *internal-combustion engines* with gear speed increasers have formed the power supply.

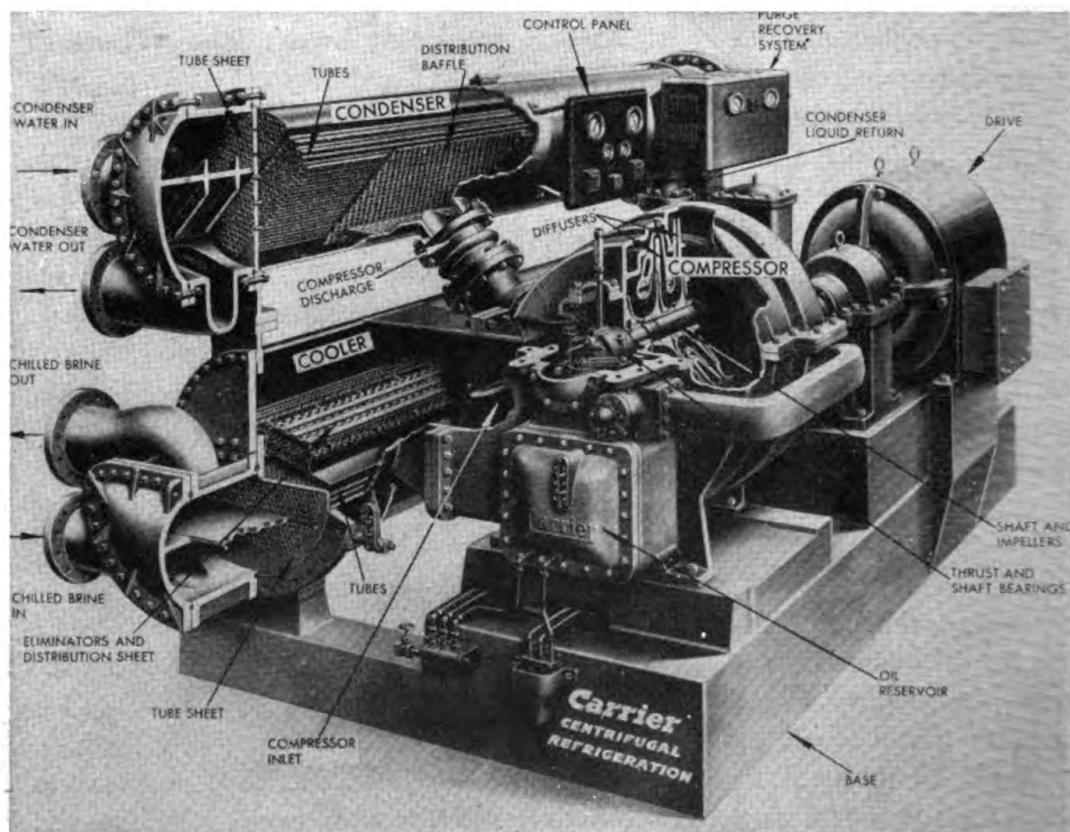


Fig. 7.22. Three-stage centrifugal refrigeration unit. Courtesy Carrier Corp.

7.13. Centrifugal-Refrigeration Equipment. Cutaway views of three typical *centrifugal* refrigeration units are shown in Figs. 7.22, 7.23, and 7.24. The unit shown in Fig. 7.22 is available in capacities ranging from 100 to 1200 tons. The *condenser*, *evaporator*, *compressor*, *speed increaser*, and *motor* are mounted on a single base. Freon-11 is used as the refrigerant, and the compressor is designed with *three stages*. The condenser is equipped with a *fully automatic purge* to eliminate air and other non-condensable gases, and the unit is equipped with an *economizer* to permit liquid intercooling and thereby increase the cycle efficiency.

The unit shown in Fig. 7.23 is similar in design to that described in the preceding paragraph and has the *condenser*, *evaporator*, *motor*, *speed increaser*, and *compressor* mounted upon a single base. The upper drum is the condenser, the lower drum the evaporator, and the small horizontal

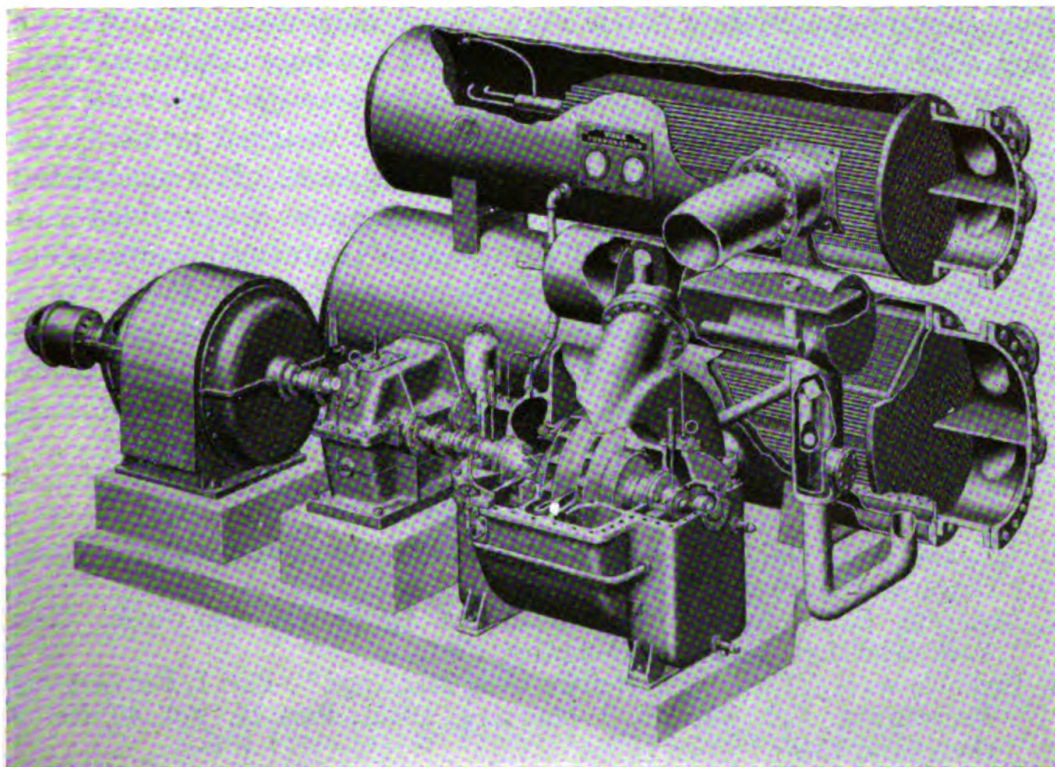


Fig. 7.23. Two-stage centrifugal refrigeration unit. Courtesy York Corp., York, Penna.

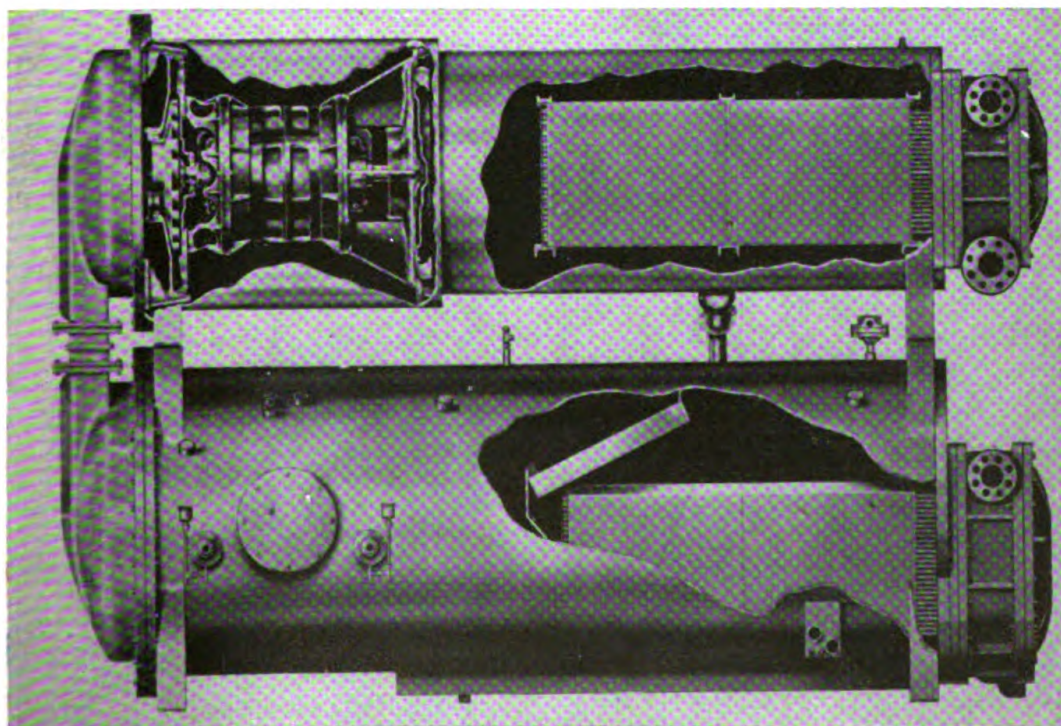


Fig. 7.24. Centrifugal refrigeration unit for water cooling. Courtesy Trane Co.

drum shown between these two is a liquid intercooler used to increase the cycle efficiency. This unit is designed with a *two-stage* compressor and, as with the unit shown in Fig. 7.22, incorporates *forced-feed* lubrication.

The centrifugal system shown in Fig. 7.24 is designed primarily for water cooling for air-conditioning and process purposes. It is constructed in sizes ranging from 50 to 200 tons. The smaller sizes are provided with *two stages* and the larger units with *four stages* of centrifugal compression, in all cases with direct electric-motor drive at 3600 rpm. Freon-113 is used as the refrigerant. The lower drum shown in Fig. 7.24 houses the *water-cooler coil*, *eliminator plates*, and, in the smaller sizes, the first *two* compression stages. The upper drum houses the *third-* and *fourth-stage* impellers and the *condenser coil*.

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PART III

***Fluid Flow, Heat Transfer, Psychometrics, and
Their Refrigeration Applications***

CHAPTER 8

Fluid Flow and Heat Transfer

8.1. Fluid Flow. Refrigeration involves heat transfer, and in most cases heat is transferred to or from one or more fluids. Therefore, in order to design or select practical, efficient, and economical heat exchangers, fluid pumps, and interconnecting piping, both as separate units and in combination in complete systems, a knowledge of the basic fundamentals of fluid flow must first be acquired. In general there are two types of flow: (1) free, natural, or gravity flow and (2) forced flow. Forced flow is further subdivided into three classifications: (a) viscous, laminar, or streamline flow, (b) turbulent flow, and (c) supersonic flow. The last does not occur as yet in common refrigeration work.

8.2. Gravity Flow. This type of flow is caused by an unbalanced difference in density of portions of a fluid that are attempting to find their equilibrium level. This unbalance may result from an increase in density of an upper portion of the fluid due to chilling or a decrease in density of a lower portion of the fluid due to heating. Chilling or heating of the fluid occurs from contact with a cool or warm substance. Hence gravity flow is closely related to heat transfer by convection, a subject that will be discussed in subsequent sections.

A common example of free flow occurs in a refrigerator using refrigerated plates or coils suspended from the ceiling. Considerable "draft" originating from these gravity-air chilling units is often perceptible. Natural flow also occurs when fluids of different densities are mixed, for example when oil and water are brought together or when water flows over a surface water cooler (surrounded by and displacing air) in modern beverage-bottling plants.

8.3. Forced Flow. When a flowing fluid is confined within a pipe and the flow is caused by mechanical rather than thermal means, the type of flow depends upon a dimensionless ratio. This expression, called the Reynolds number because of Osborne Reynold's references to it in 1883, is expressed as

$$Re = \frac{DV\rho}{\mu} \quad (8.1)$$

Experiments indicate that the critical velocity separating the two common types of unobstructed flow occurs when the Reynolds number is about 2300 or at least between 2000 and 3100. In other words, the flow is streamline for values under 2000 and is turbulent for values over 3100.

When Re is between these limits, the flow is in an unpredictable transition state. Viscosity values are given in the Appendix, Tables A.13, and A.20, and Figs. A.1, A.2, and A.3.

EXAMPLE 8.1. Using Re equal to 2300, find the critical velocities for water, saturated Freon-12 liquid, and saturated Freon-12 vapor, each at 40 F in a standard 2-in. pipe.

For water,

$$V = \frac{2300 \times (1.55 \times 0.000672)}{62.4 \times 0.172} = 0.22 \text{ ft per sec}$$

For F-12 liquid,

$$V = \frac{2300 \times (0.286 \times 0.000672)}{86.2 \times 0.172} = 0.03 \text{ ft per sec}$$

For F-12 vapor,

$$V = \frac{2300 \times (0.013 \times 0.000672)}{1.26 \times 0.172} = 0.09 \text{ ft per sec}$$

[centipoises $\times 0.000672$ = viscosity in lb per (ft)(sec)]

8.4. Streamline Flow. "Laminar," "viscous," and "straightline" are synonymous adjectives for this type of flow. They imply that all portions of the fluid move in paths parallel to the confining surface. However, this statement is not strictly true when heat is being transferred to or from the fluid, since natural convection currents are then created within the stream. In such cases the flow is classified as nonisothermal or as modified laminar flow.

The average fluid velocity at any cross section of the pipe for viscous isothermal flow is one-half the maximum velocity, which occurs in the middle of the stream. The velocity gradient across the pipe is parabolic with zero velocity at the wall. For nonisothermal flow this parabolic velocity distribution is distorted, and probably more so for a liquid than for a gas.

The pressure drop for single-phase streamline flow is usually expressed by Poiseuille's formula:

$$\Delta p = \frac{32\mu LV}{gD^2} \text{ lb per sq ft for } L \text{ ft} \quad (8.2)$$

where V is the average velocity in feet per second. The pressure drop for streamline flow may also be expressed by the Fanning equation:

$$\Delta p = \frac{4fL\rho V^2}{2gD} \text{ lb per sq ft for } L \text{ ft} \quad (8.3)$$

with $f = 16/Re$ but only when Re is under 2100. By equating equations 8.2 and 8.3, the friction factor f for streamline flow may be solved and found to equal $16/Re$.

Since viscosity and density are functions of temperature, the proper temperature must be selected. When heat is being transferred with streamline flow, McAdams¹ recommends that μ and ρ be evaluated at an average temperature t' defined as

$$t' = \frac{3t + t_w}{4} \quad (8.4)$$

where t = temperature of the main body of fluid

t_w = temperature of the pipe wall

8.5. Turbulent Flow. If the flow of a fluid is unobstructed, turbulent type of flow occurs when the Reynolds number is above 3100 and at times when it is as low as 2300. Turbulent flow is generally preferred over streamline flow when good heat transfer is an important factor. The higher the velocity, the greater the heat transfer but also the pressure drop. This means higher pumping costs so that an economic balance is involved in a consideration of heat transfer and pumping equipment.

For isothermal turbulent flow, color-band experiments have indicated that turbulence exists only in the main stream, that there is a thin laminar-flow section adjacent to the wall, and that a thin buffer zone exists between these. The average velocity over a cross section of a pipe with isothermal turbulent flow rises from 0.50 to over 0.73 of the maximum velocity as Re is increased from 2000 to 3000, then rises to 0.80 of the maximum at Re of 15,000 and remains at this ratio through Re of 90,000.²

The pressure drop for isothermal turbulent flow may be calculated from the Fanning equation:

$$\Delta p = \frac{4fL\rho V^2}{2gD} \text{ lb per sq ft for } L \text{ ft} \quad (8.3)$$

where f is the friction factor.

Analysis and tests indicate that this friction factor is a function of the Reynolds number and of the "relative roughness" or actual average roughness divided by the diameter.

Values of f for steel and wrought iron pipe are shown in Fig. 8.1.

For smooth tubes such as copper, brass, glass, etc. the following empirical equations have been proposed:

by Stoevers,³

$$f = \frac{0.0653}{(Re)^{0.228}} \quad (8.5)$$

at Re from 4000 to 1,000,000,

¹ McAdams, W. H., *Heat Transmission*, 2nd ed. New York: McGraw-Hill Book Company, Inc., 1942, p. 121.

² McAdams, W. H., *op. cit.*, pp. 101-105.

³ Stoevers, H. J., *Applied Heat Transmission*. New York: McGraw-Hill Book Company, Inc., 1941, p. 114.

and by McAdams,⁴

$$f = 0.0014 + \frac{0.125}{(Re)^{0.32}} \quad (8.6)$$

at Re from 3000 to 3,000,000.

Little information is available on friction factors for nonisothermal flow. However, McAdams recommends that the same equation and f

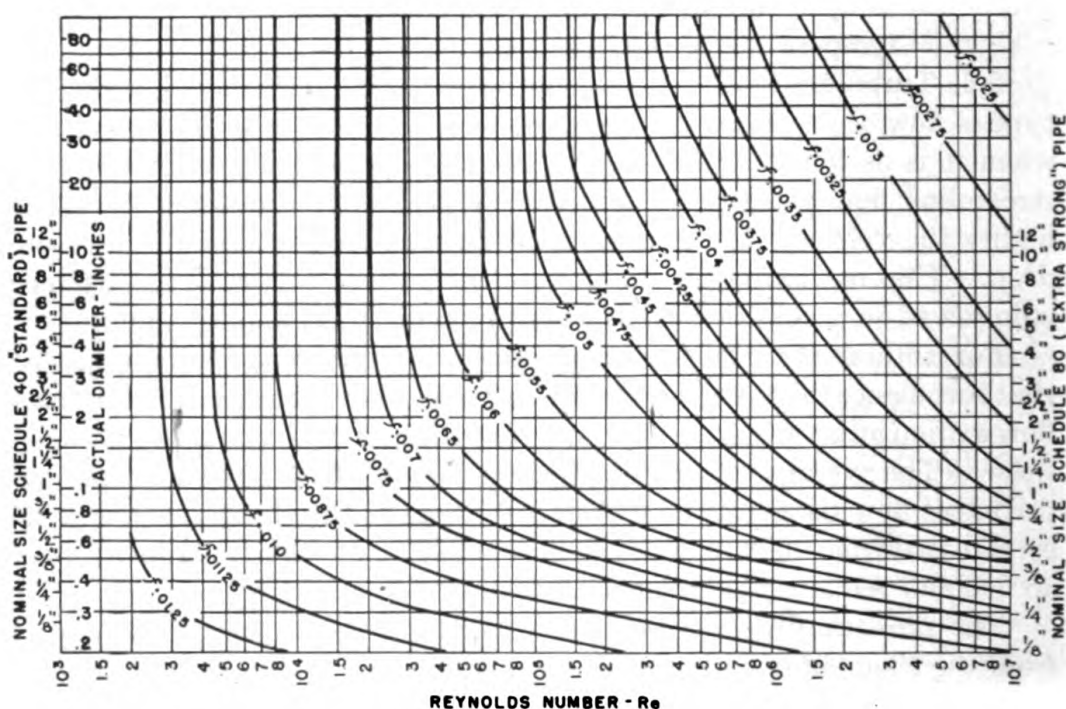


Fig. 8.1. Friction factors for turbulent flow in steel or wrought-iron pipe. From *Refrigerating Data Book*, 5th ed., 1943, p. 177. American Society of Refrigerating Engineers.

values be used but that the density and viscosity be evaluated at a temperature t_f determined by

$$t_f = \frac{t + t_w}{2} \quad (8.7)$$

where t = temperature of the main body of fluid

t_w = temperature of the pipe wall

These last two sections have dealt with single-phase flow. For two- or three-phase flow the pressure drop and heat transfer become more involved. Two-phase flow occurs in refrigeration systems when there is a significant amount of oil flowing with the refrigerant. Three-phase flow occurs in evaporators and condensers when oil is mixed with the refrigerant that is in both the liquid and vapor states.

⁴ McAdams, W. H., *op. cit.*, p. 119.

EXAMPLE 8.2. Calculate the pressure drop for 33.3 lb per minute of saturated liquid F-12 at 40 F flowing in 50 ft of $\frac{5}{8}$ -in.-O.D. type L copper tubing.

$$D = 0.625 - (2 \times 0.04) = 0.545 \text{ in.} = 0.0455 \text{ ft}$$

$$\rho = 86.2 \text{ lb per cu ft}$$

$$\mu = 0.269 \times 0.000672 = 0.000181 \text{ lb per (ft)(sec)}$$

$$V = \frac{\text{cfs}}{\text{area}} = \frac{33.3}{86.2 \times 60} \times \frac{4}{\pi \times (0.0455)^2} = 3.96 \text{ ft per sec}$$

$$Re = \frac{0.0455 \times 3.96 \times 86.2}{0.000181} = 85,800 \text{ (turbulent flow)}$$

$$f = \frac{0.0653}{(85,800)^{0.288}} = 0.0049$$

$$\Delta p = \frac{4 \times 0.0049 \times 50 \times 86.2 \times (3.96)^2}{2 \times 32.2 \times 0.0455} = 452 \text{ psf} = 3.14 \text{ psi}$$

8.6. Heat Transfer. Because this is a basic fundamental subject of considerable magnitude, several books have been written on it alone, as noted in the bibliography at the end of this chapter. Only the pertinent principles applicable to refrigeration will be discussed here.

Heat is a form of energy, the transfer of which is subject to the first and second laws of thermodynamics. All the heat lost from a source or sources must equal the heat gained by the receiver or receivers involved. When unaided by mechanical or other means, heat flow is always from the higher to the lower temperature level. Let us note how these principles apply in a simple refrigeration system.

Consider a home freezer unit holding food previously frozen. After equilibrium conditions are established, all the heat entering the box is equal to the heat absorbed by the evaporator. Heat flows from the air at about 0 F in the box to the colder evaporator. This heat is eventually discharged to the room air at about 80 F, but it is necessary to have a compressor in the system to raise the temperature level of the refrigerant. In this way the heat transferred to the refrigerant in the evaporator at a temperature below 0 F can later be transferred to the room air at 80 F passing over the condenser. The refrigerant enters the condenser at a temperature above 80 F, so that the heat here will flow of its own accord to the room air.

In the above case heat is transferred from 0 F to 80 F, but only through the aid of a refrigeration machine that requires external energy to operate it. In addition, most of this energy used to operate the refrigeration machine is also transferred to the room air at the condenser.

In some instances rapid heat transfer is desired, whereas in others a minimum of heat transfer is wanted. Again with reference to the freezer unit, if there is a rapid rate of heat transfer through the evaporator and through the condenser, there will be a reduction in the amount of heat-transfer surface area needed or in the temperature difference between

refrigerant and air, or in both. Either or both of these reductions would mean less equipment size and cost and reduced operating expense.

On the other hand, a low heat-transfer rate is desired between the room and the inside of the box. The better the resistance to heat transfer through the walls of the box, the smaller the cooling load. Such a reduction also means less cooling equipment required and lower operating costs.

Essentially, there are three methods of heat transfer, namely, conduction, convection, and radiation. This division is made in order to simplify analysis. Actually, in many cases these methods are combined. In some instances the components can be separated, but in others only the combined effect can be determined. Many unknowns still remain, and investigators are continually working in this field to increase the fund of basic knowledge.

In steady heat flow the various temperatures throughout each part of the system remain constant during heat transmission. In unsteady heat flow these temperatures vary with time.

8.7. Conduction. In this type of transmission, heat flows from the high-temperature region to the low-temperature region of a substance, transferring energy to each particle along the path with little or no apparent movement of the substance. The theory is that the hottest molecules of the substance are most active and collide with adjacent ones, imparting energy to them, and so on through to the coldest or least active molecules. Heat is conducted from the outside surface of a refrigerator wall to the cooler inside surface. In a plate-type freezer, heat flows by conduction from the inside of a piece of meat or other object to the outside and also to that part of the surface of the refrigerated plate on which the meat is resting.

The basic law for conduction of heat as presented by Fourier in 1822 may be expressed mathematically as

$$\frac{dQ}{d\tau} = -kA \frac{dt}{dx} \quad (8.8)$$

$dQ/d\tau$ is the instantaneous rate of heat flow, dQ being the amount of heat flowing in time $d\tau$ through area A taken perpendicular to the direction of heat flow. The thermal conductivity of the substance is represented by k , and $-dt/dx$ is the rate of change of temperature with respect to the distance of flow.

The thermal conductivity k is a function of temperature t , but in refrigeration a mean value of k for the temperature range involved is usually used. Then in the case of steady flow for a constant area A and length of path x , from point 1 to point 2 equation 8.8 becomes

$$\frac{Q}{\tau} \int_{x_1}^{x_2} \frac{dx}{A} = \int_{t_1}^{t_2} -k dt$$

or

$$\frac{Q}{\tau} \times \frac{(x_2 - x_1)}{A} = -k_m(t_2 - t_1)$$

and

$$q = \frac{Q}{\tau} = \frac{k_m A (t_1 - t_2)}{x} \text{ Btu per hr} \quad (8.9)$$

In many refrigeration applications the area A through which heat flows is not constant. In pipes or other cylindrical bodies of circular cross section the area perpendicular to the heat flow through the wall is $2\pi rL/12$ sq ft, where r is the radius in inches and L the cylinder length in feet.

From equation (8.8)

$$q \int_{r_1}^{r_2} \frac{dr}{2\pi rL/12} = \int_{t_1}^{t_2} -k dt$$

or

$$q \int_{r_1}^{r_2} \frac{dr}{r} = k_m(t_1 - t_2)2\pi(L/12)$$

and

$$q = \frac{k_m(t_1 - t_2)2\pi(L/12)}{\log_e \frac{r_2}{r_1}} \text{ Btu per hr} \quad (8.10)$$

In this case k is in Btu (in.) per (hr)(sq ft)(F).

By equating equations 8.9 and 8.10 with $x = r_2 - r_1$, a value for A , the correct mean area to use in equation 8.9 for cylindrical walls, is found to be

$$A_m = \frac{A_2 - A_1}{\log_e (A_2/A_1)} \quad (8.11)$$

A_m is called the log mean area for the outside surface area A_2 and the inside surface area A_1 .

An example of unsteady heat flow is that through building structures, particularly those on which sun shines. Experimental values have been obtained for the rate of heat entering a space at specific hours of the day, some of which are given in Chapter 10, Cooling-Load Calculations. Development and explanation of the theory and general equations for heat conduction in the unsteady state may be found in the references.⁵

8.8. Conductors and Insulators. Metallic substances are good heat conductors because of their crystalline structure and the presence of free electrons. This structure affords a much easier path for heat flow than

⁵ Jakob, M., and Hawkins, G. A., *Elements of Heat Transfer and Insulation*. New York: John Wiley & Sons, Inc., 1942.

McAdams, W. H., *op. cit.*

do the irregular arrangement of atoms in liquids and amorphous solids and the sparsity of molecules in gases. Cork and similar substances are good insulators because of their molecular structure and porosity or great number of air pockets. Since air is a better insulator than conductor, an insulating structure can be made by using several sheets of metal separated by air spaces. An advantage of this type of insulation is its low heat-storage capacity, making possible quicker temperature changes, which are desired for example in test chambers, than mass insulation affords. Sawdust, providing more air spaces, is a better insulator than wood. Insulation should be kept dry because water is a better heat conductor than the air that would be displaced by the wetting.

Desirable properties of low-temperature insulators are low conductivity, durability when moistened, ease of application, ability to repel vermin, and possible salvage value. To prevent moisture problems, a good vapor barrier should be installed on the side having the higher vapor pressure.

A method for calculating the most economical thickness of insulation has been proposed by Stone.⁶

$$x = \left(\frac{ak}{b} \right)^{\frac{1}{2}} - Rk$$

where x = most economical thickness

k = conductivity

b = cost per year per inch of insulation

R = sum of all heat resistances inside the structure aside from the insulation

$$a = \frac{Y(t_1 - t_2)M}{288,000}$$

Y = hours of operation per year

$(t_1 - t_2)$ = over-all temperature difference

M = cost of removing 288,000 Btu

$$b = \frac{I}{S} + I \times \frac{\text{interest rate}}{2} \times \frac{S + 1}{S}$$

I = investment per (sq ft)(in.)

S = expected life in years

EXAMPLE 8.3. Assume $t_1 - t_2 = 63 - 33 = 30$ F, $I = \$0.12$, $M = \$0.75$, $Y = 8700$, $S = 15$, $R = 2$, and $k = 0.33$. Then

$$a = \frac{8700 \times 30 \times 0.75}{288,000} = 0.68$$

$$b = \frac{0.12}{15} + 0.12 \times \frac{0.06}{2} \times \frac{15 + 1}{15} = 0.0118$$

⁶ Stone, J. F., "Cold Storage Insulation Design." *Refrigerating Engineering*, Vol. 37, No. 4 (April, 1939), pp. 229-231.

$$x = \left(\frac{0.68 \times 0.33}{0.0118} \right)^{\frac{1}{2}} - (2 \times 0.33) = 4.36 - 0.66 = 3.70 \text{ in.}$$

Therefore use 4 in., the nearest commercial thickness.

8.9. Conductivity. The units for k are usually Btu (in.) or (ft of thickness or length of path of flow) per (hr)(degree F temperature difference) (sq ft area). In using k values, the thickness unit must be noted, since all tables are not consistent.

In this text inches of thickness will be used unless stated otherwise. Methods of determining k values by test are described in the literature.⁷ These values vary with temperature, density, and character of the substance. Values for granulated cork at 2.3 lb per cubic foot vary from 0.065 at -328 F to 0.274 at 122 F. At 32 F values for cork vary from 0.22 at 3 lb to 0.38 at 20 lb per cubic foot. Cork is one of the most practical low-temperature insulators.

The commercial material having about the lowest conductivity value of any is silica aerogel, a free-flowing, voluminous solid with 90 per cent air by volume. This product has been known since about 1930, but only since World War II has it been used as a thermal insulator in home freezers and similar applications. Values of k in Btu (in.) per (hr)(sq ft)(F) with a density of about 7 lb per cubic foot are given in Table 8.1.

TABLE 8.1
CONDUCTIVITY OF SILICA AEROGEL⁸

Mean Temperature	Pressure	k
-100 F	760 mm Hg	0.105
- 50	760	0.115
0	760	0.13
+ 90	760	0.15
90	51.5	0.11
90	2.5	0.058
90	0.12	0.029

At the other extreme, at 32 F the conductivity of silver is 2928 and that of pure copper is 2712. These metals are the best conductors. For some metals the conductivity increases with temperature, and for others it decreases. Small changes in the constituents or internal structure of metals affect their conductivity values. A slight trace of arsenic in copper reduces the conductivity to one-third that of pure copper.⁹

⁷ Brown, A. I., and Marco, S. M., *Introduction to Heat Transfer*. New York: McGraw-Hill Book Company, Inc., 1942, pp. 10-11.

⁸ From Ogden, F. F., and White, John F., "The Use of Silica Aerogel as a Thermal Insulation." *Refrigerating Engineering*, Vol. 52, No. 5 (November, 1946), p. 412.

⁹ Brown, A. I., and Marco, S. M., *op. cit.*, p. 12.

Values of thermal conductivity for various solids are given in the Appendix, Tables A.14, and A.15.

Values of k for liquids, vapors, and gases are more difficult to obtain because of the radiation and convection effects. When one portion of a fluid is heated, convection currents are created unless heat is applied only at the top of the fluid. For liquids other than mercury, k varies from about 0.5 to 5.0 and changes appreciably with temperature. For gases and vapors the values lie between 0.04 and 0.3. Theory and experimental proof indicate k for gases to be a function of the viscosity and of the number of atoms in each molecule. Equations for determining the variation with temperature have been proposed.¹⁰ When these are not applicable, a straight-line variation with temperature may be used without too much error. Thermal conductivity values for liquids and for vapors are given in handbooks.

8.10. Convection Theory. The motion of the molecules in heat conduction cannot be seen or measured by ordinary means. In convection, heat is transferred by a fluid mass definitely in motion. Heat energy also flows between particles of the moving fluid or between these and a surface. Thus convection is influenced by the laws of fluid flow. Since the fluid adjacent to a surface is practically stationary, heat conduction and heat storage factors must be considered. To summarize: the shape, dimensions, temperature and character of the surface, plus the temperature, velocity, viscosity, density, specific heat, thermal conductivity, and coefficient of expansion of the fluid, and the force of gravity may affect convection heat transfer.

Because so many variables are involved, a mathematical expression for heat transfer between a surface and a fluid, like that for the friction factor of a surface, can best be developed by an application of dimensional analysis. Most texts covering convection heat transfer and friction therefore describe principles of this analysis in considerable detail and show the development of the basic equations for free and for forced convection. A brief explanation is given in the following section. For convection the units used for conductivity will be Btu (ft) per (hr)(sq ft)(F).

8.11. Dimensional Analysis. Several books have been written on this subject alone. It is not new to scientists, but its practical application to engineering problems is quite recent, principally from about 1907. The differential equations for convective heat transfer are of the most difficult type, and it was only through dimensional analysis that more simple equations and relationships were developed. Dimensional analysis makes possible the "theory of models," whereby scale test models can be used in experiments to predict the performance and improve the design of large, expensive equipment.

¹⁰ Brown, A. I., and Marco, S. M., *op. cit.*, p. 19.

This type of analysis requires care in making complete and correct assumptions, and a consistent set of units must always be used. Experiments are usually necessary to determine values of constants involved and thus complete the analysis. However, the relationship between the variables can be determined before experimenting. This information aids the experimenter in deciding which variables to hold constant and reduces the number of separate tests required.

Applications to convection heat transfer were first made by W. Nusselt, who developed the basic equations for forced convection in 1909 and for free convection in 1915.

The first step in the application of dimensional analysis is to tabulate the exponents of the dimensions of each of the factors or variables involved. The four basic dimensions used are mass, length, time, and temperature. Heat may be added if its equivalent mechanical energy is included. A general equation including a constant and all the variables raised to unknown exponents is set up. The corresponding equation of dimensions is written by substituting the corresponding dimension units for each variable, retaining the unknown exponents. Equations of the sum of the exponents for each dimension unit are then tabulated. These last equations are simplified so that all exponents are expressed in terms of a minimum of the exponents. Substitution of these exponents is made in the first general equation, and the factors are rearranged.

Let us demonstrate this method to determine the general equation for the surface coefficient with forced convection.

Variable	Symbol	Exponents of Units				
		Mass	Length	Time	Temperature	Heat
Surface coefficient.....	h_c	0	-2	-1	-1	1
Diameter.....	D	0	1	0	0	0
Velocity.....	V	0	1	-1	0	0
Absolute viscosity.....	μ	1	-1	-1	0	0
Density.....	ρ	1	-3	0	0	0
Specific heat.....	c_p	-1	0	0	-1	1
Thermal conductivity.....	k	0	-1	-1	-1	1

The exponents for h_c are obtained from another form of the basic equation of heat transfer 8.9, in which h_c , the coefficient of heat transfer by convection, is substituted for k_m/x .

$$q = h_c A (t_1 - t_2) \quad (8.12)$$

or

$$h_c = \frac{q}{A(t_1 - t_2)} = \frac{H\tau^{-1}}{L^2 T}$$

The general equation is

$$h_c = CD^a V^b \mu^c \rho^f c_p^i k^n E_k^m$$

The factor E_k , the kinetic-energy equivalent of heat, must be added to keep the units consistent. Then, substituting dimension units in the general equation,

$$HL^{-2}T^{-1}\tau^{-1} = C(L)^a(L\tau^{-1})^b(ML^{-1}\tau^{-1})^c(ML^{-2})^f(HM^{-1}T^{-1})^i \\ \times (HL^{-1}T^{-1}\tau^{-1})^m(ML^2\tau^{-2}H^{-1})^n$$

For H ,	$1 = i + m - n$
For L ,	$-2 = a + b - e - 3f - m + 2n$
For T ,	$-1 = -i - m$
For τ ,	$-1 = -b - e - m - 2n$
For M ,	$0 = e + f - i + n$

In order to obtain an equation in terms of common dimensionless numbers, the exponents must be solved, in this case, in terms of f and i .

$$\begin{aligned} a &= f - 1 & f &= f \\ b &= f & i &= i \\ e &= i - f & m &= 1 - i \\ n &= 0 \end{aligned}$$

and

$$h_c = C \frac{(D)^f}{D} (V)^f \frac{(\mu)^i}{(\mu)^f} (\rho)^f (c_p)^i \frac{k}{k^i}$$

or

$$\frac{h_c D}{k} = C \left(\frac{DV\rho}{\mu} \right)^f \left(\frac{c_{p\mu}}{k} \right)^i \quad (8.13)$$

These and other dimensionless numbers that occur frequently in heat-transfer work are identified as follows:

$$\begin{aligned} (Re) &= \frac{DV\rho}{\mu} && \text{Reynolds number} \\ (Nu) &= \frac{h_c D}{k} && \text{Nusselt number} \\ (Pr) &= \frac{c_p \mu}{k} && \text{Prandtl number} \\ (Gr) &= \frac{D^3 \rho^2 \beta g \Delta t}{\mu^2} && \text{Grashof number} \end{aligned}$$

8.12. Forced Convection. In order to evaluate the coefficient of heat transfer h_c to use in the general heat-transfer equation

$$q = h_c A (t_1 - t_2) \text{ Btu per hr} \quad (8.12)$$

and thus to determine the rate of heat flow by convection, the constant C and the exponents in equation 8.13 developed in the preceding section must be evaluated.

$$\frac{h_c D}{k} = C \left(\frac{DV\rho}{\mu} \right)^f \left(\frac{c_p \mu}{k} \right)^i \quad (8.13)$$

Numerous experiments have been made by many people using various fluids under different conditions. From these the value of the exponent for the Reynolds number has been established as 0.8, but experimental values of the exponent for the Prandtl number vary from 0.3 to 0.4 and of C from 0.023 to 0.028 for flow inside tubes. The temperature used seems to be a determining factor. The question is, should the variable be evaluated at the temperature of the main body of the fluid, at the film temperature as defined in equation 8.4 or at some other temperature?

McAdams summarized the work of many investigators and proposed the following equation with certain limitations:

$$\frac{h_c D}{k} = 0.023 \left(\frac{DV\rho}{\mu} \right)^{0.8} \left(\frac{c_p \mu}{k} \right)^{0.4} \quad (8.14)$$

When Re exceeds 10,000 and μ is greater than twice that of water, the exponent of Pr should be reduced to 0.333.

For turbulent flow in horizontal clean pipes, Brown and Marco¹¹ recommend using equation 8.14 for both heating and cooling of all fluids except for cooling fluids where $c_p \mu / k$ is greater than 10, in which case equation 8.15 is to be used.

$$Nu = 0.0265(Re)^{0.8}(Pr)^{0.3} \quad (8.15)$$

They further recommend that the properties of the fluid be evaluated at the temperature of the main body of fluid only when the temperature drop across the film is less than 10 F for liquids or less than 100 F for gases; otherwise they recommend the use of the estimated mean film temperature.

For forced flow over and perpendicular to banks of plain tubes with Re above 2000, McAdams has summarized various test data and recommends equation 8.16 for both liquids and gases flowing normal to a bank of 10 tubes staggered.

$$Nu = 0.33 F(Re)^{0.6}(Pr)^{0.333} \quad (8.16)$$

Multiply by 0.8 for tubes in line. F is taken from the following table:

No. of Tubes	2	3	4	5	6	7	8	9	10
F	0.70	0.82	0.87	0.92	0.94	0.96	0.97	0.99	1.00

¹¹ Brown, A. I., and Marco, S. M., *op. cit.*, p. 97.

The fouling of the inside of tubes reduces the heat-transfer coefficient. Gas bubbles, scale, oxides, grease, and other foreign matter in the pipe have an appreciable effect. Roughing the inside pipe surface increases turbulence and film conductance but also increases pressure drop. The insertion of turbulators or guide vanes also increases the coefficient and the friction resistance. The surface coefficient inside helically wound coils may be as much as 20 per cent greater than that in straight pipes because of the increased turbulence.

Simplified forms of equation 8.13 should be used with caution, since they usually apply to a very limited range of the fluid properties. Empirical equations for heat-transfer coefficients when the Reynolds number is less than 2100, when flow is in vertical tubes, or for flow over outside surfaces may be found in the literature. These usually apply only for specific cases, and no reliable general expressions have been developed.

The flow of fluids inside tubes or pipes in refrigeration applications, except in certain vertical liquid coolers, is usually forced flow in the turbulent region. For flow of fluids over outside surfaces, forced convection is used in some instances and free convection in others. Because of the complicated construction of finned tube coils, the outside-surface heat-transfer coefficient is most difficult or impossible to calculate. Therefore it is usually determined by running a series of tests on a coil or coils, and coil performance at other than test conditions is predicted from the test results.

EXAMPLE 8.4. Calculate the amount of heat flow from a clean standard 2-in. steel pipe at 86 F to 74 F water flowing at 300 ft per minute inside.

SOLUTION:

Evaluating at 80 F,

$$D = 2.375 - 0.308 = 2.067 \text{ in.} = 0.172 \text{ ft}$$

$$V = 300 \times 60 = 18,000 \text{ fph}$$

$$\mu = 2.08 \text{ lb per (ft)(hr)}$$

$$k = 0.355 \text{ Btu (ft) per (hr)(sq ft)(F)}$$

$$\rho = 62.17 \text{ lb per cu ft}$$

$$c_p = 0.998 \text{ Btu per lb}$$

$$Re = \frac{0.172 \times 18,000 \times 62.17}{2.08} = 92,600$$

$$Pr = \frac{0.998 \times 2.08}{0.355} = 5.85$$

$$h_c = 0.023 \frac{0.355}{0.172} (192,000)^{0.8} (5.85)^{0.4} = 915$$

$$q = 915(86 - 74) = 11,000 \text{ Btu per (hr)(sq ft of inside surface)}$$

8.13. Free Convection. The general expression for the heat-transfer coefficient with natural convection can also be developed with diensiomnal analysis. Because the variables differ somewhat, a slightly different

expression than that for forced flow is obtained. Since the velocity in free flow is due to the buoyancy force, this force acting through a distance L is equal to the kinetic energy producing the velocity V , or

$$\rho\beta \frac{\Delta t}{2} L = \frac{V^2}{2g}$$

Then $V^2 = \rho g \beta \Delta t L$.

Using L for D and substituting in equation 8.13 gives the same result as development by dimensional analysis, or

$$\frac{h_o L}{k} = C \left(\frac{g \beta \Delta t L^3 \rho^2}{\mu^2} \right)^{1/2} \left(\frac{c_p \mu}{k} \right)^i \quad (8.17)$$

Tests of free convection with various fluids and surfaces indicate that the exponents $f/2$ and i are equal. Equation 8.17 can then be written as

$$h_o = C \frac{k}{L} \left(\frac{g \beta \Delta t L^3 \rho^2 c_p}{\mu k} \right)^i \quad (8.18)$$

Tests also indicate that the exponent i depends upon the product of $Gr \times Pr$ (the fraction in parentheses) and that the constant C depends upon the shape and orientation of the surface. Table 8.2 shows values obtained for fluids on the outside of heated surfaces but can also be used for the cooling of fluids. The variables should be evaluated at the mean or average temperature of the surface and of the main body of the fluid. ($\beta = 1/T$ for perfect gases.)

TABLE 8.2
EXPONENT AND CONSTANT VALUES FOR EQ. 8.18¹²

$\frac{g\beta\Delta t L^3\rho^2c_p}{\mu k} = aL^3\Delta t$	$<10^3$	10^3-10^9		$>10^9$	
Application	i	i	C	i	C
Vertical plates.....	<0.25	0.25	0.55	0.333	0.13
Vertical cylinders.....			0.45		0.11
Horizontal cylinders.....			$0.45-0.55$		$0.11-0.13$
Horizontal plates facing up*			0.71		0.17
Horizontal plates facing down*			0.35		0.08
Spheres (L radius).....			0.63		0.15
(Note: L must not exceed 2)					

* L = Narrowest dimension.

¹² From Brown, A. I., and Marco, S. M., *Introduction to Heat Transfer*. New York: McGraw-Hill Book Company, Inc., 1942, pp. 113–114.

Simplified forms of equation 8.18 can be used for a specific fluid in a limited temperature range, in which case the value of $\frac{g\beta\rho^2c_p}{\mu k} = a$ will be constant. Then

$$h_c = \frac{Ck}{L} (\Delta t L^3 a)^{1/4} \quad (8.19)$$

In the case of vertical surfaces in air at standard pressure, and when L is less than 2 ft, as with refrigerated plates, $ka^{0.25}$ is about 0.525 at 0 F and

$$h_c = 0.29 \left(\frac{\Delta t}{L} \right)^{0.25} \text{ Btu per hr (sq ft)(F)} \quad (8.20)$$

This equation is for convection only. There is also heat transfer to the plate by radiation. An additional point of interest is that the coefficient h_c is not affected by the presence of another plate more than 1.5 in. away.

8.14. Condensing and Boiling. Two important parts of the common refrigeration system are the condenser and the evaporator. In these the refrigerant is changing state, and theoretical analysis of the process relative to heat transfer is complicated. Although extensive studies and experiments have been carried on in this field, the available information is incomplete and unsatisfactory. Factors influencing h include roughness, wettability, shape, position and newness of the surface, temperature, variation, pressure, conductivity, surface tension, viscosity, and density of the liquid and vapor, and the rate of heat flow.

There are two types of condensation, dropwise and film, and either or a combination of both may occur. Dropwise condensation coefficients are greater than film-type values; but dropwise condensation, which may occur when the surface is contaminated, is unstable. Nusselt proposed equation 8.21 which is valid for film condensation of quiescent vapors:

$$h_c = C \left(\frac{g\rho^2 h_f k^3}{L\mu\Delta t} \right)^{0.25} \quad (8.21)$$

where $C = 0.943$ for vertical surfaces

$C = 0.725$ for horizontal tubes

L = height for vertical surfaces, diameter for horizontal tubes

Coefficients for other condensation conditions and for boiling are quite unpredictable and require tests for determination. Values for water and most common refrigerants range from 100 to 1000 Btu per (hr)(sq ft)(F), but no reliable relation with the variables involved has been found. The best agreement among individual substances has been obtained by plotting the equation

$$h_c = C_1 + C_2 q \quad (8.22)$$

where C_1 and C_2 are constants

q = the rate of heat transfer per hour (square foot)

8.15. Radiation. Heat from the sun is transferred by radiation to surfaces on earth, and even through glass into a conditioned space, irrespective of whether heat is being conducted through the glass from the space to the outside or from the outside into the space. This phenomenon is utilized in the design of solar houses but is often overlooked in the design of year-round air-conditioning installations. A system may work satisfactorily in summer and in winter but still fail on cool, sunny spring days, particularly if there is considerable glass on the south side of the building. On such days the solar heat gain plus the internal heat may be greater than the heat lost by conduction to the outside, so that cooling is required in the south rooms while heating may be needed in the north rooms, especially if a north wind is blowing.

Heat is also radiated from walls and other surfaces to the colder refrigerated plate in a low-temperature space. In fact, all surfaces radiate heat to other surfaces "seen" by them and without altering the temperature of the intervening air. The higher the surface temperature, the more heat radiated, so that the net transfer is always in the direction toward the lowest temperature body.

By means of radiation, ice can be made in atmospheres at temperatures above 32 F as was done by the ancient Egyptians. If shallow trays of water are placed on the ground under a clear sky on a cool evening, enough heat will radiate to the colder interstellar space to freeze the water.

The amount of heat transferred by radiation from a perfect radiator is expressed by the Stefan-Boltzmann law, which in equation form is

$$q = 0.174 \times 10^{-8} \times T^4 \text{ Btu per (hr)(sq ft)} \quad (8.23)$$

Actually there is no perfect radiator, but the emissivity powers of all surfaces are compared with that of the theoretical or so-called "black body." The radiant heat absorptivity power of a surface is numerically equal to its emissivity power in accordance with Kirchhoff's law, derived in 1859. Since reflection is the opposite of absorption, the better a surface is as a reflector, the poorer it is as a radiator.

Oxygen, nitrogen, hydrogen, and chlorine do not absorb radiation, so that the net heat radiated from surface 1 to surface 2 separated by any of these mediums may then be expressed as

$$q = 0.174 F_r F_A A \left[\left(\frac{T_1}{100} \right)^4 - \left(\frac{T_2}{100} \right)^4 \right] \text{ Btu per hr} \quad (8.24)$$

where F_r = a function of the emissivities of the two surfaces and is affected by their configuration.

F_A = a function of the configuration or orientation, the separating distance, and the areas of the surfaces.

Fortunately, $F_A = 1$ for the common applications. Values of F_e for common applications are given in Table 8.3. Emissivity or absorptivity factors depend somewhat on the temperature of the surface. Color is effective only in the case of very high temperatures or solar radiation. It may be of advantage to wear white instead of dark clothing when out in the sun but not when inside a hot boiler room. Recent tests indicate that the type of dye or coloring matter may be more influential than the actual shade or color. Emissivity factors for various surfaces are given in hand books.

TABLE 8.3
EMISSIVITY FUNCTIONS

Surface Relationships	F_e
Infinite parallel planes or completely enclosed body, large compared with enclosing body	$\frac{1}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1}$
Completely enclosed body, small compared with enclosing body	ϵ_1
Concentric spheres or infinite cylinders (subscript 1 for enclosed body)	$\frac{1}{\frac{1}{\epsilon_1} + \frac{A_1}{A_2} \left(\frac{1}{\epsilon_2} - 1 \right)}$

The hydrocarbons, water vapor, CO_2 , SO_2 , HCl , and NH_3 , absorb and emit radiation enough to warrant consideration. The general equation for radiant transfer between a gas and a surface is

$$q = \epsilon_s A_s (I_g - I_s) \text{ Btu per hr} \quad (8.25)$$

I_g and I_s are intensity factors of the gas and of the surface and depend upon the partial pressure of the gas, the length of path in question, and the respective temperatures.

8.16. Combined Heat Transfer. In actual practice the various principles of heat transfer must be combined in order to solve many problems. In insulated walls and pipes, convection occurs at the inner and outer surfaces, radiation may occur at either or both surfaces, and conduction occurs between the surfaces.

When heat is transferred from a fluid to a surface, through the material, and then from the other surface to the fluid or space on the other side, the following equations can be developed and applied.

The general equation is

$$q = UA\Delta t \text{ Btu per hr} \quad (8.26)$$

For a composite wall as shown in Fig. 8.2,

$$U = \frac{1}{\frac{1}{f_o} + \frac{1}{C} + \frac{x_1}{k_1} + \frac{1}{a} + \frac{x_2}{k_2} + \frac{1}{f_i}} \quad (8.27)$$

where

f_i, f_o = inside and outside surface coefficients accounting for convection and radiation

k_1, k_2 = conductivities of materials 1 and 2

x_1, x_2 = thicknesses of materials 1 and 2

C = conductance of non-homogenous material from surface to surface

a = heat transmission from surface to surface across an air space by conduction, convection, and radiation.

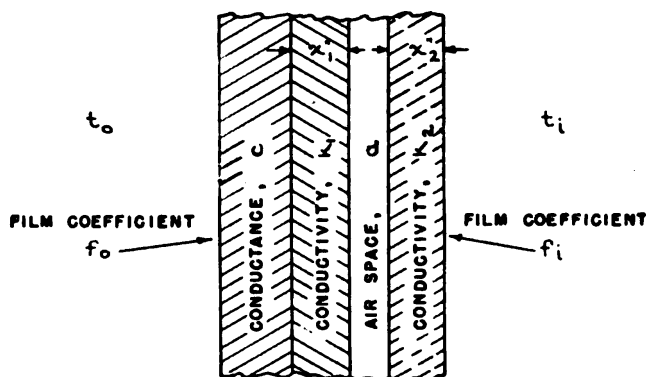


Fig. 8.2. Composite wall section.

The inside-surface area involved is substituted for A , and Δt is the difference between inside and outside air temperatures.

The surface temperature when desired may be calculated by first calculating the rate of heat transfer q from the air on one side of the wall to the air on the other side of the wall. Then using this q and the same area in equation 8.28 solve for t_s .

$$q = fA(t_i - t_s) \text{ Btu per hr} \quad (8.28)$$

t_i is the temperature of the adjacent air, and f is the surface coefficient, which is 1.65 for still air, as in Example 8.5(a).

When equation 8.26 is applied to a section of an insulated pipe as shown in Fig. 8.3,

$$U = \frac{1}{\frac{1}{h_c} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{1}{f_o}} \quad (8.29)$$

The log mean area (see equation 8.11), must be substituted for A . Δt is the temperature difference between the fluid inside and the fluid outside.

In the case of a heat exchanger where the temperature of one or both of the fluids varies from inlet to outlet, Δt should be the log mean tem-

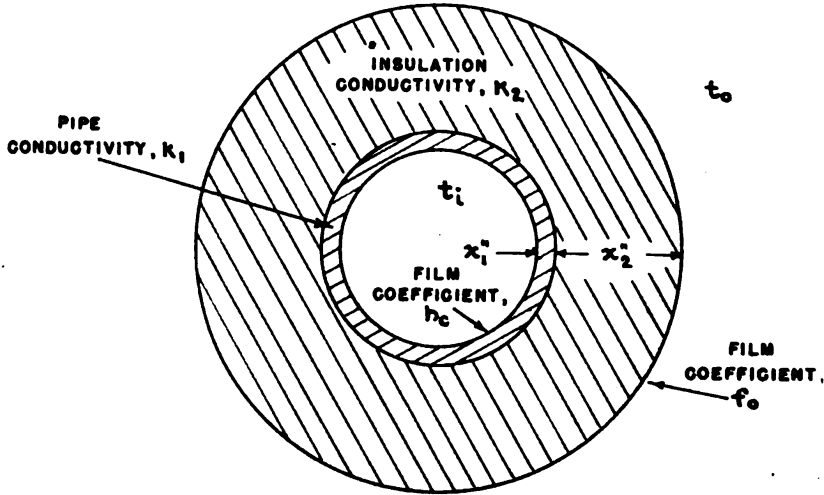


Fig. 8.3. Insulated pipe section.

perature difference if the entire unit is under consideration. As illustrated in Fig. 8.4 this would be

$$\Delta t_m = \frac{\Delta t_1 - \Delta t_2}{\log_e \frac{\Delta t_1}{\Delta t_2}} \quad (8.30)$$

Since counterflow of the fluids will produce a greater Δt_m than parallel flow, the former should always be used for economic reasons if practical.

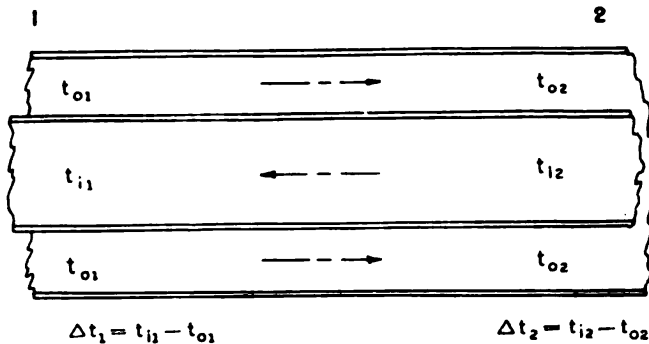


Fig. 8.4. Double-pipe heat exchanger.

EXAMPLE 8.5. (a) Consider a refrigerated space kept at 0 F. The walls are of 6-in. cork with $\frac{1}{2}$ -in. cement plaster on each side. The temperature outside the box is 80 F. Determine the rate of heat gain through the wall and the inside wall temperature if $f_1 = 1.65$.

$$q = UA(t_o - t_i)$$

$$U = \frac{1}{\frac{1}{1.65} + \frac{0.5}{12} + \frac{6}{27} + \frac{0.5}{12} + \frac{1}{1.65}} = \frac{1}{0.606 + 0.042 + 22.222 + 0.042 + 0.606}$$

$$U = 0.043$$

$$q = 0.043 \times 1 \times (80 - 0) = 3.44 \text{ Btu per (hr)(sq ft)}$$

$$q = f_i A (t_s - t_i)$$

$$3.44 = 1.65 \times 1 \times (t_s - 0)$$

$$t_s = 2.08 \text{ F}$$

(b) Calculate the theoretical surface heat transfer for a 12-in. vertical refrigerated plate with a surface temperature of -15 F located in the room. $\epsilon = 0.9$ for enamel on metal.

$$h = h_c + h_r$$

$$h_c = 0.29 \left[\frac{0.0 - (-15)}{1.0} \right]^{0.25} = 0.29 \times 1.97 = 0.57$$

$$q_r = 0.174 \times 1 \times 0.9[(4.617)^4 - (4.446)^4] = 10.02$$

$$h_r = \frac{Q_r}{\Delta t} = \frac{10.02}{17.08} = 0.59$$

$$h = 0.57 + 0.59 = 1.16 \text{ Btu per (hr)(sq ft)(F)}$$

8.17. Series and Parallel Heat Conduction. In practical applications of heat conduction the heat may flow through the substances involved by series or parallel paths or by a combination of both. An analysis of heat conduction in this respect is similar to that for the conduction of electricity through series and parallel circuits. In a series heat-flow path, the heat resistances of the materials involved are additive, or

$$R_T = R_1 + R_2 + R_3 \dots \quad (8.31)$$

Since conductance is the reciprocal of resistance,

$$\frac{1}{C_T} = \frac{1}{C_1} + \frac{1}{C_2} + \frac{1}{C_3} \dots \quad (8.32)$$

or for area A ,

$$\frac{1}{C_T} = \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} \dots \quad (8.33)$$

When heat is conducted through parallel paths, the conductances are additive, or

$$C_T = C_1 + C_2 + C_3 \dots \quad (8.34)$$

For an area A consisting of $A_1 + A_2 + A_3 \dots$

$$C_T = \frac{k_1 A_1}{x_1} + \frac{k_2 A_2}{x_2} + \frac{k_3 A_3}{x_3} \dots \quad (8.35)$$

If C_T is desired on a per unit area basis, A_1 , A_2 , A_3 , and so on, are taken as percentages of the total area and expressed as decimals in equation 8.35. The above principles are applied in the following example.

EXAMPLE 8.6. Calculate the over-all coefficient U for a double metal wall separated by pine 2×4 's placed 12 in. on centers if the space between the 2×4 's contains a fill-type insulation having a conductivity of 0.3.

SOLUTION. Neglecting the resistance of the metal in this case, the conductance from the inside metal surface to the outside metal surface is

$$\begin{aligned}
 C &= \frac{k_1 A_1}{x_1} + \frac{k_2 A_2}{x_2} \\
 &= \frac{0.8 \times 1.625/12}{3.625} + \frac{0.3 \times 10.375/12}{3.625} \\
 &= 0.0298 + 0.0715 = 0.1013 \text{ Btu per (hr)(sq ft)(F)}; \\
 U &= \frac{1}{\frac{1}{f_1} + \frac{1}{C} + \frac{1}{f_2}} = \frac{1}{\frac{1}{1.65} + \frac{1}{0.1013} + \frac{1}{1.65}} = 0.0905 \text{ Btu per (hr)(sq ft)(F)}
 \end{aligned}$$

Serious thought should be given to the type of material used for separation of the inner and outer shell and for the door edges on low-temperature structures. If high-conductivity materials such as metals are used, condensation may occur on the warm side where the separators or door edges come in contact with it. Heat will be conducted so rapidly that the surface at such places may be below the dew point of the adjacent air.

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PROBLEMS

8.1. Calculate the velocity for saturated ammonia vapor at 40 F in a 2-in. pipe for a Reynolds number of 2300.

8.2. Calculate the pressure drop in 100 ft of 2-in. steel pipe for saturated ammonia vapor at 40 F if the velocity is (a) 0.47 ft per second, (b) 47 ft per second.

8.3. (a) Calculate the pressure drop of 40 F saturated Freon-12 vapor flowing through 100 ft of 2 $\frac{1}{8}$ -in.-O.D. copper pipe from a 10-ton evaporator. Liquid refrigerant enters the expansion valve at 80 F. (b) Calculate the pressure drop for the same conditions except for 0 F saturated F-12 vapor. (c) Calculate the pressure drop for the same conditions as in (a) but for 1 $\frac{5}{8}$ -in.-O.D. pipe.

8.4. Determine the Btu per hour heat loss from 100 ft of 2 $\frac{1}{8}$ -in.-O.D. copper pipe insulated with $\frac{1}{2}$ in. of hair felt if the room temperature is 100 F and the exterior surface of the pipe is at 40 F.

8.5. Calculate the rate of heat loss from a $1\frac{1}{2}$ -in. steel pipe carrying a 25 per cent calcium chloride brine at 8 F if the pipe is at 12 F and the velocity is 300 fpm.

8.6. (a) What is the rate of heat flow by convection from a $1\frac{1}{2}$ -in. steel pipe at 12 F in air at 26 F? (b) What is the rate of heat flow by radiation if the surrounding walls of the large room in which the pipe is located are 32 F? Emissivity of the pipe is 0.63.

8.7. Calculate the heat loss from a $1\frac{1}{2}$ -in. steel pipe insulated with 2 in. of cork if the inside film coefficient is 900, the inside fluid temperature is 8 F, the surrounding outside temperature is 26 F, and the outside film coefficient is 1.5. k for the pipe is 310.

8.8. Calculate the total heat transfer from the top and bottom of a 36×12 -in. horizontal plate at 20 F in a room at 40 F.

8.9. How much heat-transfer surface area is required for a double-pipe heat exchanger that is to cool liquid F-12 from 80 F to 75 F by the saturated vapor leaving a 10-ton evaporator at 40 F? ($U = 100$.)

8.10. Calculate the over-all coefficient of heat transfer for a double metal wall separated $2\frac{5}{8}$ in. by wood spacers $1\frac{5}{8} \times 2\frac{5}{8}$ in., 16 in. on centers, if the remainder of the space is filled with glass wool at 2.5 lb per cubic foot. Neglect the thermal resistance of the metal.

8.11. (a) Determine the outside surface temperature opposite the wood in Problem 8.10 if the inside still-air temperature is 0 F and the outside still-air temperature is 80 F. (b) Determine the outside surface temperature opposite the glass wool for the same conditions.

8.12. Calculate U for an outside wall constructed of 8 in. of concrete, 3 in. of corkboard, and $\frac{1}{2}$ in. of cement plaster.

8.13. (a) Calculate U for a roof of slate, 1 in. wood sheathing, 2×4 -in. rafter 18 in. on centers, and 4 in. of commercial-grade glass wool. A $\frac{1}{2}$ -in. thick insulating board is under the rafters. (b) What is U if 6 in. of glass wool is used with 2×6 -in. rafters?

8.14. (a) Calculate U for an inside self-supporting partition consisting of $\frac{1}{2}$ in. of cement plaster, 2 in. of corkboard, another $\frac{1}{2}$ in. of cement plaster, another 2 in. of corkboard, and then another $\frac{1}{2}$ in. of plaster. (b) Determine U if 3 in. of corkboard is substituted for one of the 2-in. sections. (c) Determine U if 3 in. of corkboard is substituted for both 2-in. sections.

8.15. (a) A cabinet wall is made of two sheets of 0.04-in. thick galvanized iron separated by $3 \times \frac{1}{8}$ -in. steel channel irons placed every 12 in. Calculate U if the 3-in. space is filled with commercial glass wool. (b) Calculate U if 1×3 -in. pine spacers are substituted for the channel irons.

8.16. Calculate the outside surface temperature opposite the channel irons in Problem 8.15(a) if the outside air temperature is 80 F and the temperature inside the cabinet is 0 F.

8.17. (a) Calculate U for a partition consisting of $\frac{1}{2}$ -in. plaster on each side of $1\frac{5}{8} \times 3\frac{5}{8}$ -in. pine studs 12 in. on centers and with the $3\frac{5}{8}$ -in. wide space between the studs filled with sawdust. (b) Calculate U if the studs are 18 in. on centers.

CHAPTER 9

Psychrometrics

9.1. Significance. "Psychrometric" literally means "pertaining to the measurement of cold," but in modern usage it refers to the state of the atmosphere with reference to moisture. The earth's atmosphere—the air we breathe and the air in most refrigerated spaces—is mainly a mixture of dry air and water vapor. Of course in special cases other gases or vapors might also be present. But generally speaking, in refrigeration applications the atmosphere is considered a mixture of dry air and water vapor.

Although the relative amount of water vapor in the atmosphere is very small (about 0.08 per cent by weight at 0 F saturated and only 1.56 per cent at 70 F saturated), variations in this amount as well as changes in temperature are very significant and important in air conditioning. Relative to human comfort, there is much truth in the common expression, "It's not the heat, it's the humidity." Humidity means the amount of water vapor within a given space.

In many instances humidity control is very important. Poor register is often noticed in colored printing, particularly in the Sunday comics; adjacent colors overlap or do not meet. A change in atmospheric humidity between the printing of the different colors probably was the cause, since paper shrinks and stretches as the humidity changes. In a meat cooler, too low a humidity can cause shrinkage, or drying out, which means a loss of product and less income to the owner. On the other hand, too high a humidity promotes a growth of mold and possible spoilage of the product. Therefore it should be evident that a study of psychrometry, including the properties of air-vapor mixtures and their measurement and control, is important in air conditioning.

9.2. Definitions. Before proceeding, let us define some of the expressions used with reference to air-vapor mixtures. The following are taken from the *Heating Ventilating Air Conditioning Guide*.¹

Dry-bulb temperature—The temperature of a gas or mixture of gases indicated by an accurate thermometer after correction for radiation.

Wet-bulb temperature—Thermodynamic wet-bulb temperature is the temperature at which liquid or solid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature. Wet-bulb temperature

¹ *Heating Ventilating Air Conditioning Guide*, 1948. American Society of Heating and Ventilating Engineers, New York, Chapter 1.

(without qualification) is the temperature indicated by a wet-bulb psychrometer constructed and used according to specifications.

Dew point temperature—The temperature at which the condensation of water vapor in a space begins for a given state of humidity and pressure as the temperature of the vapor is reduced. The temperature corresponding to saturation (100% relative humidity) for a given absolute humidity at constant pressure.

Relative humidity—The ratio of the actual partial pressure of the water vapor in a space to the saturation pressure of pure water at the same temperature.

Humidity ratio—In a mixture of water vapor and air, the weight of water vapor per pound of dry air. Also called Specific Humidity.

Absolute humidity—The weight of water vapor per unit volume, pounds per cubic foot or grams per cubic centimeter.

Saturated air—A mixture of dry air and saturated water vapor, all at the same dry-bulb temperature.

Per cent saturation—The ratio of the weight of a given volume of water vapor to the weight of an equal volume of saturated water vapor at the same temperature.

Psychrometer—An instrument for ascertaining the humidity or hygrometric state of the atmosphere.

9.3. Properties of Air-Vapor Mixtures. Atmospheric air has properties like those of its separate constituents. The ones of greatest interest for a given specific mixture in air conditioning include pressure, temperature, volume, density, ratio of water to air quantity, and enthalpy. The pressure, temperature, and volume, or density, of a vapor are interdependent. As learned in thermodynamics, this relationship for a perfect gas can be expressed by the perfect-gas equation, namely,

$$pV = wRT, \quad \text{or} \quad p = \rho RT, \quad \text{or} \quad pv = RT \quad (9.1)$$

Actually, there is no perfect gas, but many of the commonly used gases approach the ideal. The perfect-gas equation can be applied in solving most problems involving air and many gases. When greater accuracy is required, as in calculating values for moist-air tables, an expanded form of the following equation has been proposed:²

$$pv = RT - A(T)p \quad (9.2)$$

where $A(T)$ = an involved coefficient expressed as a function of absolute temperature and taking into account the intermolecular forces

Vapors do not follow the perfect-gas laws, and the perfect-gas equation cannot be applied to them. This is the main distinction between a vapor and a gas. For this reason tables or charts must be used to determine the properties of vapors such as steam and the common refrigerants. However, under certain extreme conditions some vapors come near to

² Goff, John A., and Gratch, S., "Thermodynamic Properties of Moist Air." *Heating, Piping and Air Conditioning: ASHVE Journal Section*, Vol. 17, p. 334 (June, 1945).

being perfect gases, and then the gas equation can be used with fair accuracy. For example, water vapor or steam which is very highly superheated or which is at very low pressures can be treated as a gas with small error. The water vapor in the atmosphere in refrigeration work is usually at a low partial pressure (0.01 to 0.5 psi). Therefore, for normal accuracy the perfect-gas equation can be applied to both the moisture and the dry air that exist in the common atmosphere.

9.4. Pressure. Dalton's law states that the pressure of a mixture of gases is equal to the sum of the partial pressures of each constituent and that each constituent occupies the entire volume. However, precise measurements made during the last few years indicate that this law as well as Boyle's and Charles' laws are only approximately correct. Modern tables of atmospheric-air properties are based on the corrected versions. For most commercial problems sufficient accuracy can be obtained by applying the proper basic equation and using values from the modern tables.

Several equations for calculating the partial pressure of the water vapor in the air have been proposed. Dr. Carrier's equation, presented in 1911, is probably most widely used in ordinary problems when charts are not available:³

$$p_H = p_{wb} - \frac{(p - p_{wb})(t_{db} - t_{wb})}{2800 - 1.3t_{wb}} \quad (9.3)$$

When values from the latest steam tables are used, the constants change slightly so that

$$p_H = p_{wb} - \frac{(p - p_{wb})(t_{db} - t_{wb})}{2830 - 1.44t_{wb}} \quad (9.4)$$

At temperatures below 32 F this equation applies only for temperatures of and vapor pressures over supercooled water. For partial pressures of water vapor over ice the denominator becomes $3160 - 0.09t_{wb}$; and p_{wb} must be the partial pressure of vapor over ice at t_{wb} , the temperature of an iced wet bulb.

The partial pressure of the dry air is taken as the difference between the total or barometric pressure and the partial pressure of the water vapor.

9.5. Volume and Density. Volume of a gas or mixture refers to the space occupied by a given quantity. Density usually means weight per unit volume. In air-conditioning work additional relationships are found useful, such as weight of water vapor in an air-vapor mixture per unit weight of dry air in the mixture. This relationship at times has been called "humidity ratio," "specific humidity," and "absolute humidity."

³ Carrier, W. H., "Rational Psychrometric Formulae." *Transactions A.S.M.E.*, Vol. 33 (1911), p. 1005.

The American Society of Heating and Ventilating Engineers is attempting to standardize on *humidity ratio*. The weight of water vapor is often expressed in grains (7000 grains = 1 lb) and the weight of dry air in pounds. Another useful relationship is the cubic feet of mixture per pound of dry air.

Note that in these last two expressions the reference factor is pounds of dry air. This reference makes these properties useful in air-conditioning problems because the total weight of dry air entering an air-conditioning unit in a given time is the same as the weight of dry air leaving the unit in the same time. The rate of flow of the air-vapor mixture, normally expressed in pounds per hour, into the unit would not be the same as the rate of mixture flow out of the unit if any moisture were added to or removed from the mixture within the unit. Because of a change in density, the cubic feet per minute of dry air and of the mixture on leaving a unit may be different from the cubic feet per minute of either on entering the unit. But the pounds of dry air entering and leaving in a given time always remain the same after steady-flow conditions are established.

9.6. Temperature. Three different temperatures are used when referring to an air-vapor mixture, namely, *dry-bulb*, *wet-bulb*, and *dew-point* temperatures. The dry-bulb temperature is usually measured with a common thermometer or thermocouple. The wet-bulb temperature is measured with a common thermometer or thermocouple having the bulb or junction completely covered by a water-moistened cloth or wick. Moisture evaporates from the wick into the surrounding air, and before equilibrium is reached, a portion of the heat used is taken from the thermometer bulb or thermocouple wires, as the case may be, thus lowering the temperature indicated below the indicated dry-bulb temperature. The drier the surrounding air, the greater the rate of evaporation and the lower the wet-bulb temperature indicated.

As moisture from the wet bulb evaporates, the air surrounding the bulb becomes more humid, causing the rate of evaporation to decrease and the indicated wet-bulb temperature to rise. Therefore, in order to measure the wet-bulb temperature of air in a given space, a continuous sample of this air must pass around the wet bulb. Tests have been conducted to determine the effect of varying air velocities over thermocouple and thermometer wet bulbs.⁴ These indicate that velocities from 500 to 1000 fpm give a minimum error for the conventional-size thermometer bulb at temperatures from 20 to 60 F. The temperatures of near-by surfaces can affect the wet bulb reading due to radiation, and varying air velocities also influence this effect. Furthermore, the size of the wet bulb is significant in that the smaller the diameter, the lower the air velocity required. Thus, obtaining an accurate wet-bulb reading can become rather com-

⁴ Wile, D. D., "Psychrometry in the Frost Zone." *Refrigerating Engineering*, Vol. 48, No. 4 (October, 1944), p. 291.

plicated. The application of the reading determines the accuracy required. Soft fine-mesh cotton tubing is recommended for a wick; it should cover the bulb plus about an inch of the stem. The wick should be watched and replaced before it becomes dirty or crusty. Using distilled water is recommended to give greater accuracy longer.

The dew-point temperature is the temperature of an air-vapor mixture at which moisture will start to condense out of the air as the air is cooled; or it is the temperature of a surface just as condensate starts to collect on the surface when the surface is being cooled. In other words, it is the saturation temperature of water at the partial pressure of the water vapor in the air-vapor mixture. Therefore, if the partial pressure of the vapor is known, the dew point can be determined directly from steam tables. If the dry-bulb and wet-bulb temperatures are known, the partial pressure can be found from equation 9.4. Also, the dew-point temperature can be found on a psychrometric chart by reading the temperature where a constant moisture line from the dry-bulb and wet-bulb point intersects the saturation curve.

The terms "wet-bulb depression" and "dew-point depression" are often used. The first refers to the difference in degrees Fahrenheit between the dry-bulb temperature and the wet-bulb temperature. Similarly, the second refers to the difference between the dry-bulb and dew-point temperatures.

9.7. Humidity. The student of air conditioning should learn the difference between relative humidity and per cent saturation (see §9.2). By definition, relative humidity is expressed as

$$H_R = \frac{p_H}{p_s} \quad (9.5)$$

where p_H = actual partial pressure of the water vapor

p_s = saturation pressure of water vapor at the dry-bulb temperature

Per cent saturation is expressed as

$$\phi = \frac{w_H}{w_s} \quad (9.6)$$

where w_H = humidity ratio

w_s = humidity ratio at saturation at the dry-bulb temperature

The relationship between the two can be expressed as

$$\phi = \frac{(p - p_s)}{(p - p_H)} \times H_R \quad (9.7)$$

where p = total barometric pressure

By applying Dalton's law and the perfect-gas law and neglecting the intermolecular forces, an expression for calculating the ratio of water vapor to dry air in the atmosphere can be derived as follows:

$$w_w = \frac{p_H v}{R_w T} = \frac{p_H v}{85.6 T}$$

But

$$v = \frac{w_{da} R_{da} T}{p_{da}} = \frac{w_{da} \times 53.3 \times T}{p_{da}}$$

\therefore

$$\frac{w_w}{w_{da}} = \frac{p_H \times 53.3 \times T}{p_{da} \times 85.6 \times T} = 0.622 \frac{p_H}{p_{da}}$$

or

$$w_H = \frac{0.622 p_H}{p - p_H} \text{ lb water vapor per pound of dry air} \quad (9.8)$$

and

$$w_H = \frac{4354 p_H}{p - p_H} \text{ grains water vapor per pound of dry air} \quad (9.9)$$

where the subscripts used are

H and w for water vapor
 da for dry air

9.8. Heat Content. In dealing with gases, vapors, and air-vapor mixtures the expression *enthalpy* is often used as indicative of the heat content, since heat transferred at constant pressure equals enthalpy change. Most air-conditioning processes are practically at constant pressure. The enthalpy of an air-vapor mixture is equal to the enthalpy of the dry air plus the enthalpy of the water vapor, taken in proportion to the weight of each present. For convenience in air-conditioning problems this enthalpy is expressed as Btu of mixture per pound of dry air. Of course values of heat content are not absolute, nor do they indicate the total heat energy involved. The various expressions for heat content merely indicate the difference in heat energy between a fixed datum or reference condition and the condition under consideration. The datum generally taken for steam and water properties is that for saturated liquid at 32 F. The datum for dry air is 0 F and standard atmospheric pressure.

The enthalpy of an air-vapor mixture can be calculated from the following equation:

$$h = c_p t_{db} + w_H h_v \quad (9.10)$$

where $h_v = h_g$ at dew-point temperature $+ 0.45 (t_{db} - t_{dp})$ or, approximately, $h_v = 1061 + 0.444 t_{db}$, based on the latest steam tables. Also

t_{db} = dry-bulb temperature

t_{dp} = dew-point temperature

Therefore, without using a steam table, enthalpy can be calculated approximately from

$$h = c_p t_{db} + w_H(1061 + 0.444 \times t_{db}) \quad (9.11)$$

Enthalpy values at standard pressure only are given in the table of moist-air properties (A.18) in the Appendix. Using this table, to determine the enthalpy of a partly saturated mixture at 29.92 in. of mercury the following equation can be applied at temperatures below 150 F:

$$h = h_a + \phi h_{as} \quad (9.12)$$

where h_a and h_{as} are taken from Table A.18.

Equations 9.10, 9.11, and 9.12 may be applied at temperatures below freezing as well as above 32 F. The amount of heat transferred to or from air at constant pressure, the process applicable in most cases, is equal to the difference in enthalpies at the initial and final conditions.

Other terms for heat content have been formulated that are said to save time and eliminate confusion with some sacrifice in accuracy. One of these expressions is called *total heat* and defined by the following equation:

$$T.H. = c_p t_{wb} + w_{Hwb} h_{gwb} \quad (9.13)$$

where t_{wb} = wet-bulb temperature

w_{Hwb} = humidity ratio of saturated air at the wet-bulb temperature

h_{gwb} = enthalpy of saturated steam at the wet-bulb temperature

This equation applies above and below freezing. Another expression called *sigma heat content* is defined as follows:

$$h\Sigma = h - w_H h_{fwb} \quad (9.14)$$

$$h\Sigma = c_p t_{wb} + w_{Hwb} h_{fgwb} \quad (9.15)$$

where h_{fwb} = approximately $(t_{wb} - 32)$

h_{fgwb} = approximately $(1093 - 0.556 t_{wb})$

These last two equations also apply above and below freezing but only for vapor over water. For vapor over ice the following equations apply:

$$h\Sigma = h - w_H h_{iwb} \quad (9.16)$$

$$\text{or} \quad h\Sigma = c_p t_{wb} + w_{Hwb}(h_{gwb} - h_{iwb}) \quad (9.17)$$

where

$$h_{iwb} = \text{approximately } (0.48 t_{wb} - 159)$$

$$(h_{gwb} - h_{iwb}) = \text{approximately } (1220 - 0.036 t_{wb})$$

Note that in equations 9.13, 9.15, and 9.17 the heat content is a function of the wet-bulb temperature. For this reason many psychrometric charts indicate total heat or sigma heat content values opposite the lines of constant wet-bulb temperature.

Also note that at saturation, when the dry-bulb and the wet-bulb temperatures are the same, the total heat and the enthalpy values are the same. The sigma heat content is equal to this same value minus the heat

of liquid of the moisture present. Or, as indicated by equations 9.14 and 9.16, the sigma heat content is always equal to the enthalpy value minus the heat of liquid at the wet-bulb temperature.

To summarize the comparison of these three heat content expressions, the difference in enthalpies indicates the actual amount of heat added or taken away at constant pressure, but lines of constant enthalpy on a psychrometric chart would only add confusion. Lines of constant total heat parallel the wet-bulb lines, but differences in total heats are not true indications of the amount of heat added or removed. Total heat values do not differ greatly from enthalpy values at high relative humidities, but there is no simple relationship between them. Lines of constant sigma heat contents parallel the wet-bulb lines. Differences in sigma heat

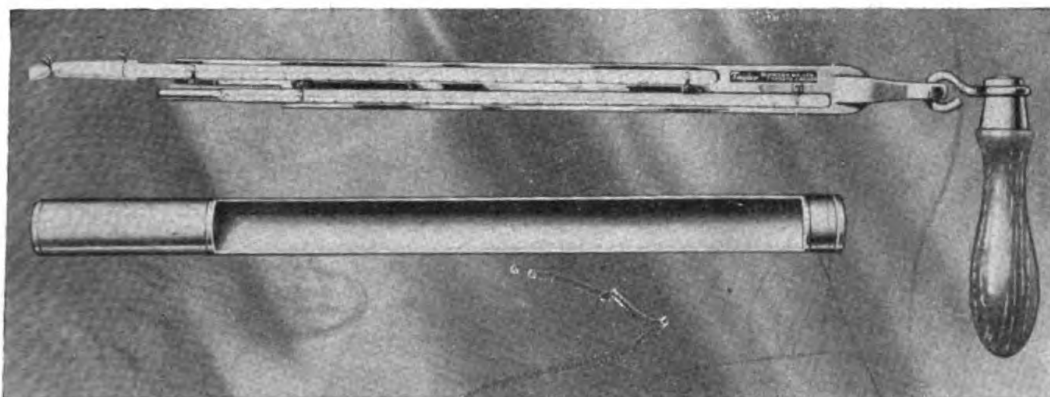


Fig. 9.1. Sling psychrometer with 10 in. thermometers. Courtesy Taylor Instrument Co.

contents are not true indications of the amount of heat exchanged, but there is a simple difference between enthalpy and sigma heat content.

9.9. Psychrometers. Numerous types of instruments are made for measuring psychrometric properties of air. The most commonly used types employ two thermometers, one dry-bulb and one wet-bulb. As previously stated, a wet bulb in still air will give an erroneous reading. This reading will be 10 to 20 per cent of the wet-bulb depression below the temperature of adiabatic saturation for a radiation-shielded wet bulb, and it will be 10 to 20 per cent of the wet-bulb depression above the temperature of adiabatic saturation if unshielded. The error is about 18 per cent of the wet-bulb depression at 60 F. Radiation to and from surfaces having temperatures considerably different from that of the air sample should be avoided. The sling psychrometer, a common type of which is shown in Fig. 9.1, is the type most generally used, especially for checking conditions on the job. The instrument is rotated by hand to get air movement across the bulbs. The same effect can be accomplished with stationary bulbs by drawing air across the bulbs with a hand-operated aspirator (see Fig. 9.2) or by a small blower like those used in

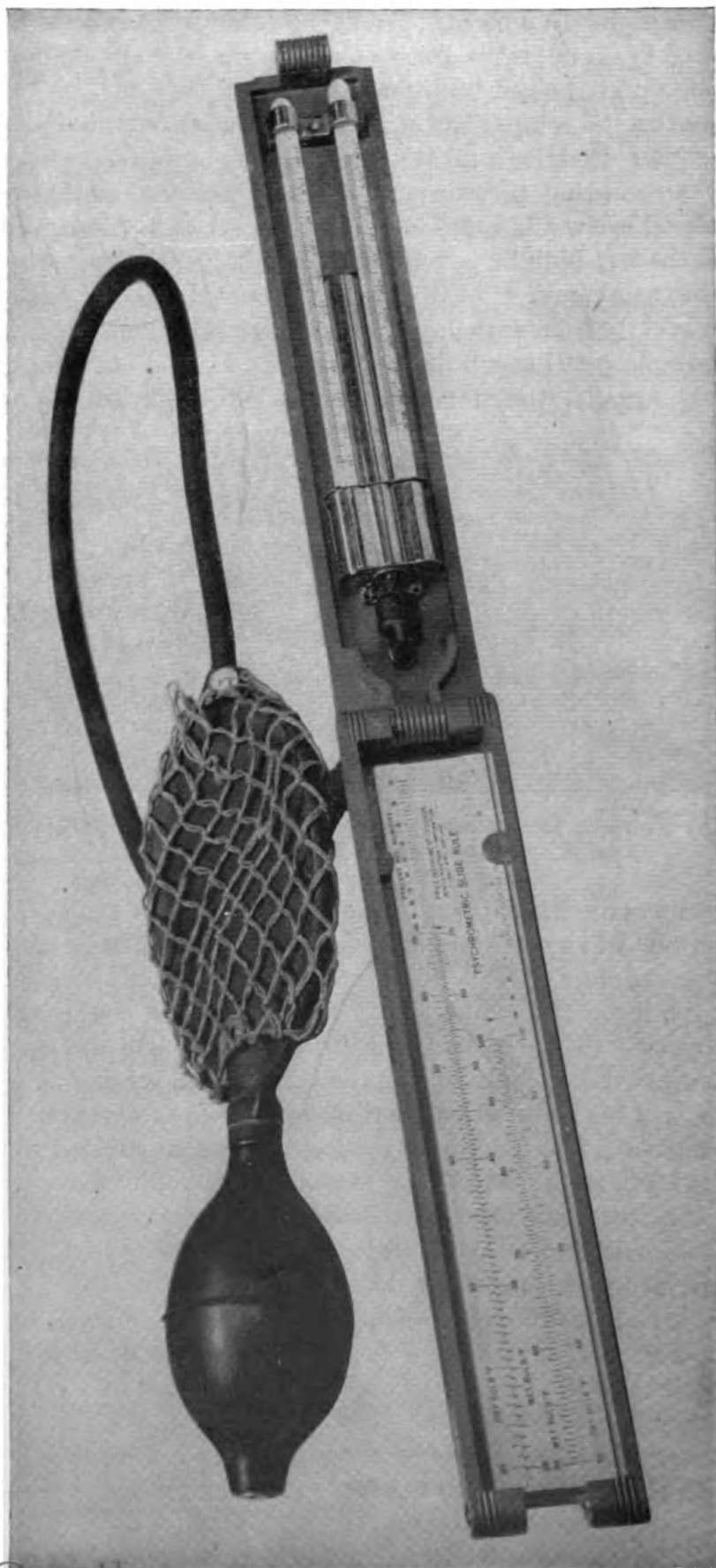


Fig. 9.2. Aspiration psychrometer. Courtesy Friez Instrument Division, Bendix Aviation Corp.

portable hair driers. The latter type requires less work from the operator but needs a source of electricity.

Many humidity-control instruments and indicators employ human hairs, which are sensitive to relative humidity but must be kept clean and in calibration. Certain other hygroscopic materials are sometimes used to indicate or control relative humidity.

One of the most accurate types of psychrometric instruments measures the dew-point temperature.⁵ A thin plate of highly polished metal is used with a controlled cooling fluid flowing behind it and a thermocouple fastened just behind the polished surface. As the fluid temperature is lowered and raised, the polished surface is observed with a long-focus microscope. The plate temperatures are recorded when moisture just appears and disappears on the plate as the air sample passes over it. These readings are averaged to obtain the dew-point temperature.

EXAMPLE 9.1. The following are applications of the preceding basic information to specific problems.

(a) A sling psychrometer gives readings of 80 F dry bulb and 67 F wet bulb. Calculate the relative humidity indicated if the barometer is 29.92 in. Hg.

From the steam tables, saturation pressure at 80 F = 1.0323 and at 67 F = 0.6668 in. Hg. Using Eq. 9.4,

$$\begin{aligned} p_H &= p_{wb} - \frac{(p - p_{wb})(t_{db} - t_{wb})}{2830 - 1.44t_{wb}} \\ &= 0.6668 \frac{(29.92 - 0.67)(80 - 67)}{2830 - 1.44 \times 67} \\ &= 0.6668 - 0.1392 = 0.5276 \text{ in. Hg} \end{aligned}$$

Using equation 9.5,

$$H_R = \frac{p_H}{p_s} = \frac{0.5276}{1.0323} = 51 \% \text{ relative humidity}$$

Calculate the per cent saturation.

Using equation 9.7,

$$\begin{aligned} \phi &= \frac{(p - p_s)}{(p - p_H)} H_R \\ &= \frac{(29.92 - 1.03)}{(29.92 - 0.53)} \times 0.51 = 50.1 \% \text{ saturation} \end{aligned}$$

Dew-point temperature from steam tables is the saturation temperature at 0.5276 in. Hg = 60.3 F.

Calculate the humidity ratio and volume per pound of dry air. Using equation 9.8,

$$w_H = \frac{0.622p_H}{p - p_H} = \frac{0.622 \times 0.5276}{29.92 - 0.53} = 0.01119 \text{ lb water vapor per lb dry air}$$

⁵ Ewell, A. W., "Thermometry in Hygrometric Measurements." *Refrigerating Engineering*, Vol. 40, No. 1 (July, 1940), p. 27.

$$v = \frac{wRT}{p} = \frac{0.01119 \times 85.6 \times 539.6}{0.5276 \times 0.491 \times 144} = 13.85 \text{ cu ft per lb dry air}$$

Calculate the enthalpy, total heat, and sigma heat content.

Using equation 9.11 and c_p at 80 F = 0.2404,

$$\begin{aligned} h &= c_p t_{db} + w_H(1061 + 0.444 \times t_{db}) \\ &= 0.2404 \times 80 + 0.01119(1061 + 0.444 \times 80) = 31.50 \text{ Btu per lb dry air} \end{aligned}$$

Using equation 9.12 (since pressure is 29.92 in. Hg),

$$\begin{aligned} h &= h_a + \phi h_{as} \\ &= 19.22 + 0.501 \times 24.47 = 31.50 \text{ Btu per lb dry air} \end{aligned}$$

Using equation 9.13,

$$\begin{aligned} T.H. &= c_p t_{wb} + w_H h_{gwb} \\ &= 0.2403 \times 67 + 0.01424 \times 1090.5 = 31.62 \text{ Btu per lb dry air} \end{aligned}$$

Using equation 9.14,

$$\begin{aligned} h\Sigma &= h - w_H h_{fwb} \\ &= 31.50 - 0.01119 \times 35 = 31.11 \text{ Btu per lb dry air} \end{aligned}$$

Using equation 9.15,

$$\begin{aligned} h\Sigma &= c_p t_{wb} + w_H h_{fgwb} \\ &= 0.2403 \times 67 + 0.01424 \times 1055.4 = 31.12 \text{ Btu per lb dry air} \end{aligned}$$

(b) Calculate the same values as above at the same temperatures 80 F dry bulb, 67 F wet bulb, but at a pressure of 24 in. Hg (equivalent to about 5900 ft elevation).

$$\begin{aligned} p_H &= 0.6668 - \frac{(24.00 - 0.67)(80 - 67)}{(2850 - 1.44 \times 67)} \\ &= 0.6668 - 0.1110 = 0.5558 \text{ in. Hg} \end{aligned}$$

$$H_R = \frac{0.5558}{1.0323} = 53.9\% \text{ rel. hum.}$$

$$\phi = \frac{(24 - 1.03)}{(24 - 0.56)} \times 0.539 = 52.6\% \text{ saturation}$$

Dew-point temp. = 61.8 F

$$w_H = 0.622 \times \frac{0.5558}{24.00 - 0.56} = 0.01475 \text{ lb per lb dry air}$$

$$v = \frac{0.01475 \times 85.6 \times 539.6}{0.5558 \times 0.491 \times 144} = 17.3 \text{ cu ft per lb dry air}$$

$$h = 0.2404 \times 80 + 0.01475(1061 + 0.444 \times 80) = 35.40 \text{ Btu per lb dry air}$$

$$T.H. = 0.2403 \times 67 + 6.22 \frac{0.6668}{24.0 - 0.67} \times 1090.5 = 35.52 \text{ Btu per lb dry air}$$

$$h\Sigma = 35.40 - 0.01477 \times 35 = 34.88 \text{ Btu per lb dry air}$$

$$\text{or } h\Sigma = 0.2403 \times 67 + 0.0178 \times 1055.4 = 34.88 \text{ Btu per lb dry air}$$

9.10. Subfreezing Conditions. Psychrometry below 32 F requires extra precautions if much accuracy is desired. First, since the maximum wet-bulb depression is only 18 F and diminishes as the temperatures decrease, errors in thermometer readings become more significant. For example, a thermometer error of 0.1 F at 60 F means only 0.5 per cent error in relative humidity, whereas 0.1 F at -10 F means 4.3 per cent error in relative humidity. Matched calibrated thermometers should be used, and the lower the temperature, the finer the graduations required; 0.1 F graduations or finer are recommended near 0 F.

The second precaution is to allow sufficient time for equilibrium. The wick must be completely covered with ice, and a longer time is required to reach equilibrium than at higher temperatures. More reliable results at dry bulbs under 32 F can be obtained without a wick and by coating the bulb with ice. This can be done by alternately dipping the bulb in ice water, then holding it in freezing air.

The third precaution, if a wick is used, is to make certain whether it is covered with ice or whether it contains supercooled water. Water on a wick can be chilled to the low 20's and not always freeze. Errors over 6 per cent in relative humidity are possible at 32 F wet bulb if the wrong vapor-pressure values are used. The vapor pressures at saturation over ice are less than those over supercooled water. However, at partial saturation the vapor pressure, humidity ratio, and therefore the enthalpy of an air-vapor mixture over ice may be less or greater than at the same temperatures over water. In using charts or tables it should be noted whether values are for air over water or over ice, and the proper ones should be chosen.

The fourth precaution is to avoid the radiation of heat from the observer and from other surfaces that are above dry-bulb temperature.

The fifth precaution of having the proper air velocity over the bulbs applies at low temperatures as well as at other temperatures.

When using an iced wet bulb, it is possible to obtain a wet-bulb reading more than 0.6 F higher than the dry-bulb reading. This phenomenon has been observed at temperatures as low as -22 F.⁶ It occurs if the air contains supersaturated vapor when considered over ice but is saturated or only partly so when considered over supercooled water. The vapor upon contacting the iced bulb condenses and freezes. In so doing, heat is given up, which raises the indicated wet-bulb temperature.

For a given air-vapor mixture with a dew-point temperature below 32 F there is only one dry-bulb temperature, but the other properties have two values, one for over ice and the other for over water.

Methods of calculating the properties at subfreezing conditions are given under their respective sections earlier in this chapter.

⁶ Ewell, A. W., *loc. cit.*

EXAMPLE 9.2. (a) For a barometer reading of 29.92 in. of mercury calculate the humidity ratio over water and over ice at 30 F dry bulb and 20 F wet bulb.

Over water,

$$\begin{aligned}
 p_{ws} &= 0.1096 \text{ in. Hg} \\
 p_H &= 0.1096 - \frac{(29.92 - 0.11)(30 - 20)}{2830 - 1.3 \times 20} \\
 &= 0.1096 - 0.1070 = 0.0026 \text{ in. Hg} \\
 w_H &= \frac{4354 \times 0.0026}{29.92 - 0.0026} = 0.38 \text{ grain per lb dry air}
 \end{aligned}$$

Over ice,

$$\begin{aligned}
 p_H &= 0.1028 - \frac{(29.92 - 0.10)(30 - 20)}{3160 - 0.09 \times 20} \\
 &= 0.1027 - 0.0943 = 0.0084 \text{ in. Hg} \\
 w_H &= \frac{4354 \times 0.0084}{29.92 - 0.0084} = 1.23 \text{ grains per lb dry air}
 \end{aligned}$$

(b) Calculate the same as in (a) but for 21 F dry bulb and 20 F wet bulb.

Over water,

$$\begin{aligned}
 p_H &= 0.1096 - \frac{(29.92 - 0.11)(21 - 20)}{2830 - 1.3 \times 20} \\
 &= 0.1096 - 0.0107 = 0.0989 \text{ in. Hg} \\
 w_H &= \frac{4354 \times 0.0989}{29.92 - 0.0989} = 14.50 \text{ grains per lb dry air}
 \end{aligned}$$

Over ice,

$$\begin{aligned}
 p_H &= 0.1028 - \frac{(29.92 - 0.10)(21 - 20)}{3160 - 0.09 \times 20} \\
 &= 0.1027 - 0.0094 = 0.0933 \text{ in. Hg} \\
 w_H &= \frac{4354 \times 0.0933}{29.92 - 0.0933} = 13.62 \text{ grains per lb dry air}
 \end{aligned}$$

Note that in (a) the humidity ratio is less over water than over ice, whereas at the temperatures in (b) the opposite is true. In (a) at 30 F dry bulb a relative humidity of $\frac{0.0034}{0.1645}$, or 2.0 per cent, produces the same wet-bulb reading using supercooled water on the wick as a relative humidity of $\frac{0.0082}{0.1645}$, or 5.0 per cent, produces using an iced wet bulb. In (b) a relative humidity of $\frac{0.0990}{0.1078}$, or 86.5 per cent, indicates the same wet bulb using water on the wick as a relative humidity of $\frac{0.0933}{0.1078}$, or 92.0 per cent, indicates on an iced wet bulb. These relative humidity values are based on saturation over ice, which is standard practice in this country. English and German values of relative humidity are based on saturation over water and therefore are lower for the same temperatures.⁷

⁷ Ewell, A. W., "Humidity Measurement." *Refrigerating Engineering*, Vol. 35, No. 3 (March, 1938), p. 176.

(c) Calculate the enthalpies and sigma heat contents for the conditions in (a) and (b).

At 30 F dry bulb and 20 F wet bulb:

Over water,

$$h = 0.24 \times 30 + \frac{0.38}{7000} \times (1061 + 0.444 \times 30) = 7.26 \text{ Btu per lb dry air}$$

$$h\Sigma = 0.24 \times 20 + \frac{1.6}{7000} \times (1093 - 0.556 \times 20) = 7.28$$

Over ice,

$$h = 0.24 \times 30 + \frac{1.23}{7000} \times (1061 + 0.444 \times 30) = 7.39$$

$$h\Sigma = 0.24 \times 20 + \frac{1.5}{7000} \times (1220 - 0.036 \times 20) = 7.41$$

At 21 F dry bulb and 20 F wet bulb:

Over water,

$$h = 0.24 \times 21 + \frac{14.5}{7000} \times (1061 + 0.444 \times 21) = 7.26$$

$$h\Sigma = 0.24 \times 20 + \frac{1.6}{7000} \times (1093 - 0.556 \times 20) = 7.28$$

Over ice,

$$h = 0.24 \times 21 + \frac{13.62}{7000} \times (1061 + 0.444 \times 21) = 7.12$$

$$h\Sigma = 0.24 \times 20 + \frac{1.5}{7000} \times (1220 - 0.036 \times 20) = 7.41$$

9.11. Psychrometric Charts. In order to conserve time in determining properties of air-vapor mixtures, charts of various types have been designed. One presented in 1946 and copyrighted by the Carrier Corporation, has, like some preceding ones, dry-bulb temperature as the abscissa and humidity ratio as the ordinate.⁸ Copies for two temperature ranges are in the envelope on the back cover. Enthalpy values of saturated air are indicated on the extended diagonal wet-bulb lines. Lines of constant deviation from enthalpy at saturation are drawn through the region of partly saturated air. Thus the enthalpy at any condition in the range of the chart can be easily computed.

These charts are for the standard barometric pressure of 29.92 in. of mercury. A table of corrections is included to apply for values at other pressures. Values at temperatures below freezing are for over ice. In 1944 Wile presented a chart from -10 to 50 F showing values both over subcooled water and over ice at temperatures below freezing.⁹ This chart indicates sigma heat content values.

The standard barometric chart gives values of suitable accuracy for most commercial and industrial applications throughout the United States

⁸ Palmatier, E. P., and Wile, D. D., "A New Psychrometric Chart." *Refrigerating Engineering*, Vol. 52 (July, 1946), p. 31.

⁹ Wile, D. D., "Psychrometry in the Frost Zone." *Refrigerating Engineering*, Vol. 48, No. 4 (October, 1944), p. 291.

with the exception of Denver and other smaller communities at significant altitudes. Two charts, one for -50 to 60 F and the other for 40 to 140 F, at pressures from 2 to 200 psia, have been presented by Kanig.¹⁰

9.12. Processes Involving Air-Vapor Mixtures. The common processes involved in air conditioning are at practically constant pressure and can be illustrated by lines on a psychrometric chart.

(A) *Sensible Heating.* This process involves heating without changing the moisture content (humidity ratio) of the air. It therefore must be a horizontal line from left to right on the chart, as shown in Fig. 9.3. An application would be air in a duct passing over a heater. The relative humidity is lowered.

(B) *Sensible Cooling.* This process involves cooling at constant moisture content or humidity ratio of the air. It can be shown as a

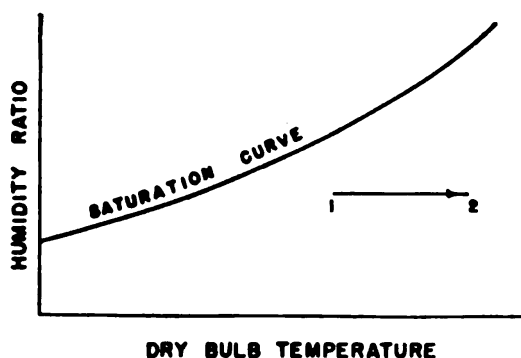


Fig. 9.3. Sensible heating.

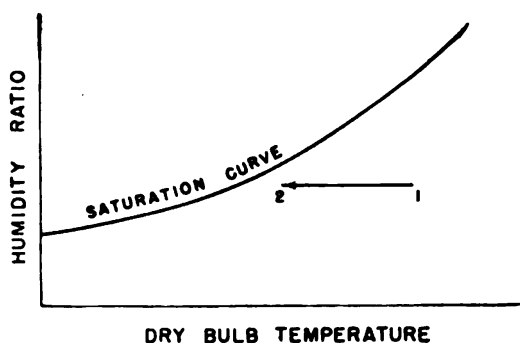


Fig. 9.4. Sensible cooling.

horizontal line from right to left on the chart as indicated in Fig. 9.4. The final temperature cannot be below the initial dew-point temperature. An example would be air in a duct passing over a cooling coil having a surface temperature above the dew-point temperature of the air. The relative humidity is raised.

(C) *Adiabatic Saturation.* "Adiabatic" means "no heat added or removed externally." "Saturation" implies adding moisture. Recirculated water is sprayed into the air, and part of it evaporates in trying to saturate the air. If the spraying is effective enough, the air leaves the spray 100 per cent saturated. Then, after equilibrium is established, the air will leave at the temperature of its entering wet bulb, and the recirculated water will be at this same temperature. This process is indicated on the chart as a line parallel to the wet-bulb lines from the entering-air condition to the saturation curve, as shown in Fig. 9.5. In commercial practice the process of evaporative cooling closely approaches this ideal process. Evaporative cooling is effective in hot dry climates when the cooler humid condition is more desirable than the hot dry condition. It

¹⁰ Kanig, H. Edmund, "Psychrometric Charts for High Altitude Calculations." *Refrigerating Engineering*, Vol. 52, No. 5 (November, 1946), pp. 434-435.

should be remembered that the leaving wet-bulb temperature cannot be lower than the spray-water temperature. The relative humidity is raised.

(D) *Cooling and Dehumidifying.* This process implies lowering both the air temperature and the humidity ratio. The actual process is difficult to show on a chart. It would depend on the type of equipment used, but in general it can be indicated by a dotted curved line, as shown in

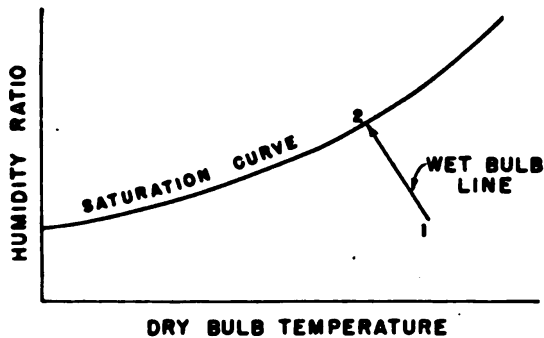


Fig. 9.5. Adiabatic saturation.

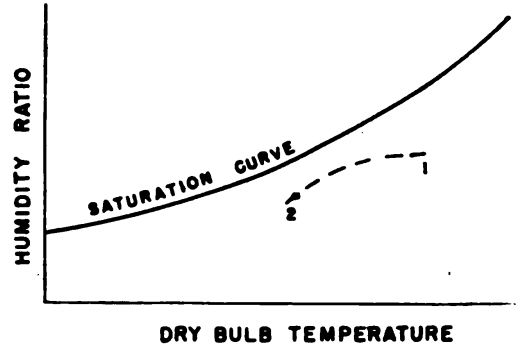


Fig. 9.6. Cooling and dehumidifying.

Fig. 9.6. This is the process commonly used in summer air conditioning in which air passes over a cold coil or through a cold water spray. The air is chilled below its dew point, thus condensing out moisture. The final relative humidity generally is higher than at the start.

(E) *Heating and Humidifying.* In this process, the reverse of the last one described, the temperature and humidity ratio both rise. On a

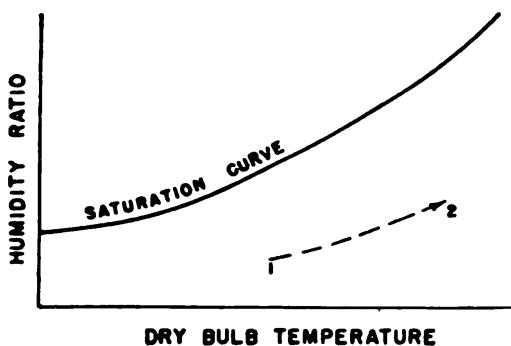


Fig. 9.7. Heating and humidifying.

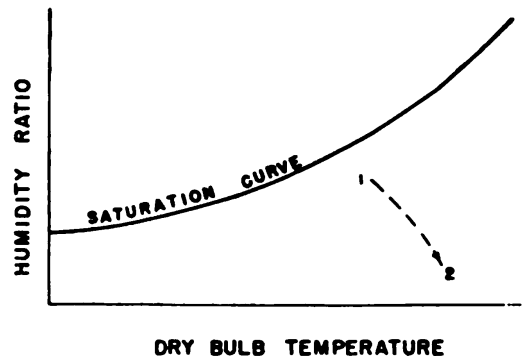


Fig. 9.8. Chemical dehumidifying.

chart it is indicated by a dotted curved line between the initial and final conditions, as shown in Fig. 9.7. This process is used in the winter to warm and humidify the air. It should be noted that the final relative humidity can be lower or higher than the initial value.

(F) *Chemical Dehumidifying.* Air can be dehumidified by passing it over chemicals that have an affinity for moisture. Usually in so doing the moisture is condensed and gives up its latent heat, raising the dry-bulb temperature of the air. The air leaves drier but warmer. The process can be indicated by a dotted line on the chart, as shown in Fig. 9.8. The

wet bulb may increase or decrease. Units employing such chemicals are used in some comfort air-conditioning installations but mainly for industrial air conditioning. Since the leaving-air temperature is usually higher than wanted, it is necessary to add a sensible cooling process to get the desired final-air condition.

(G) *Mixing Air.* When air at one condition is mixed with air at another condition, the condition of the final mixture can be indicated on a psychrometric chart by a point on the line joining the points indicating the original conditions, as shown in Fig. 9.9. The exact location of the

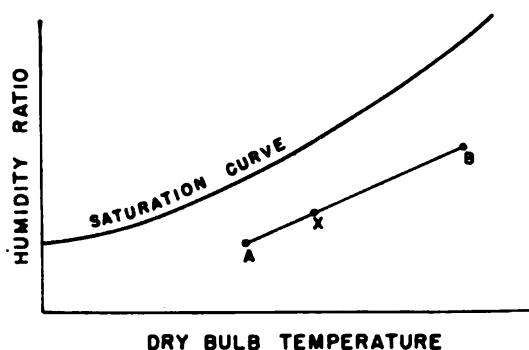


Fig. 9.9. Mixing of air A and air B.

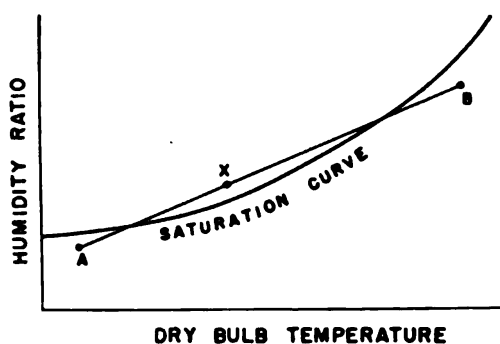


FIG. 9.10. Air mixing with final condition in fog region.

final point will depend upon the relative weights of dry air in the original air quantities. The ratio of AX to AB is equal to the ratio of the weight of dry air at B to the total weight of dry air involved.

9.13. Fogged Air. Under certain conditions saturated air can hold additional moisture in the form of minute water droplets. The area to the left or above the saturation curve on the psychrometric chart used here represents conditions of fogged air. Such an atmospheric condition can be created in more than one manner. When warm humid air is mixed with cold air, the resulting mixture will be a fog if the state of the final mixture lies in the fog region of the psychrometric chart. Figure 9.10 illustrates this condition where equal amounts of air at conditions A and B are mixed. The final condition will be represented by x . The temperature of the fog is that of the extended wet-bulb line passing through x .

Fog can also result when steam or a very fine water spray is injected into air in a quantity greater than that necessary to saturate the air. Lesser quantities incompletely mixed with the air can result in part fog and part unsaturated air.

Three methods of clearing fog are heating the fog, mixing the fog with warmer unsaturated air, and mechanically separating the water droplets from the air.¹¹

¹¹ Goodman, William, "The Psychrometric Chart, Its Application and Theory." *Heating, Piping and Air Conditioning*, Vol. 11, No. 11 (November, 1939), p. 671.

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PROBLEMS

✓ 9.1. (a) Calculate the water-vapor pressure of the atmosphere if the dry-bulb temperature is 72 F and the wet-bulb temperature is 50 F at standard barometer. Calculate (b) the relative humidity, (c) the per cent saturation, (d) the humidity ratio, (e) the enthalpy of the mixture, (f) the total heat, and (g) the sigma heat content.

9.2. For air at 25 F dry bulb and 20 F wet bulb over ice, calculate at 29.92 in. Hg (a) relative humidity, (b) per cent saturation, (c) humidity ratio, (d) enthalpy, (e) total heat, and (f) sigma heat content.

9.3. Calculate the values in Problem 9.2 if the barometer is 25 in. Hg.

9.4. How many Btu per hour are required to heat 1000 cfm of air at 25 F dry bulb and 20 F wet bulb to 70 F dry bulb and 50 F wet bulb? How many gpm of water must be evaporated?

9.5. Determine the dry-bulb and wet-bulb temperatures of a mixture of 1000 cfm of air at 95 F dry bulb and 78 F wet bulb and 3000 cfm of air at 75 F dry bulb and 65 F wet bulb.

9.6. What will be the dry-bulb temperature if air at 100 F and 20 per cent relative humidity is saturated adiabatically to 90 per cent relative humidity?

9.7. What would be the final dry-bulb, wet-bulb, and humidity condition if 100 cfm of air at 75 F dry bulb and 70 F wet bulb mixes with 100 cfm of air at 32 F dry bulb and 90 per cent relative humidity?

9.8. (a) Determine the degree or per cent saturation of water vapor in air at 70 F dry bulb and 50 F wet bulb. (b) What is the relative humidity? (Barometer reads 29 in. Hg.)

9.9. Calculate the cubic feet of air-vapor mixture per pound of dry air at 51 F and 80 per cent relative humidity.

9.10. How many tons of refrigeration are required to cool 1000 cfm of air from 95 F dry bulb, 78 F wet bulb to 80 F dry bulb and 50 per cent relative humidity?

9.11. For air at 0 F dry bulb and 40 per cent relative humidity, calculate over ice (a) wet bulb, (b) dew point, (c) humidity ratio, and (d) enthalpy. (Barometer is 30 in. Hg.)

9.12. Air at 35 F dry bulb and 90 per cent relative humidity is cooled to 25 F. Determine the amount of moisture removed and the total enthalpy given up per pound of dry air.

9.13. A room is to be maintained at 40 F and 80 per cent relative humidity. Air is to be supplied at 30 F to absorb 100,000 Btu per hour sensible heat and 35 lb of moisture per hour. How many pounds of dry air per hour is required? What should be the dew-point temperature and relative humidity of the supply air?

9.14. Can an air condition of 70 F and 80 per cent be obtained by evaporative cooling and reheat from a condition of (a) 100 F and 12 per cent, (b) 95 F and 18 per cent, (c) 105 F and 10 per cent? Diagram each process on a schematic psychrometric chart.

9.15. How high a relative humidity must 95 F air have in order to obtain saturated air at 65 F by evaporative cooling?

CHAPTER 10

Cooling-Load Calculations

10.1. Types of Loads. The refrigerating engineer may encounter a wide variety of applications in his practice. In order to determine the proper equipment to use in each case he must first calculate the load and study its characteristics. Some of the more common types of loads include those for comfort air conditioning, industrial air conditioning, product chilling, product freezing, product storage, chilling water, milk, oil or other fluids, liquefying gases, and freezing liquids. Many of the fundamentals of heat transmission are applied in load calculations. In most cases heat transfer by conduction occurs, and in many instances heat transfer by convection or by radiation, or by both, takes place.

The components of the heat gain in an air-cooled space are of two types, sensible heat and latent heat. It is important to differentiate between the two, since the ratio of the total quantity of one to that of the other affects the determination of the proper balance of equipment required. The air quantity, coil surface area, refrigerant temperature, and the sensible-latent heat ratio must balance for a given set of room conditions. Since sensible heat manifests itself by a temperature change, any heat source which raises the dry-bulb temperature of the air in a conditioned space causes a sensible-heat gain. Latent heat is the heat transferred during a change of state of a substance, such as the latent heat of evaporation or the latent heat of condensation. As air passes through conditioning equipment and moisture is condensed, latent heat is given up. This air must enter the conditioned space with a humidity ratio low enough to produce the desired room conditions after absorbing the undesired moisture within the space. Thus any source that adds water vapor to the air in a conditioned space causes a latent-heat gain. The components that make up the totals of each of these heat gains are explained and discussed in the following sections.

10.2. Design Conditions for Air Cooling. Since several of the items in the total-heat gain for a space are functions of the difference between inside and outside temperature and humidity, the proper design conditions must be selected. In comfort conditioning it is desirable, as the term implies, to maintain conditions that are comfortable to the majority of the occupants most of the time. From numerous tests conducted with human "guinea pigs" to determine desirable room conditions, the American Society of Heating and Ventilating Engineers developed their Effective Temperature Charts.

Effective temperature is an empirical index of warmth that takes into account the combined effect of temperature, humidity, and air movement. Any conditions for any one effective temperature produce the same sensation of warmth to most human beings as is felt in an atmosphere of

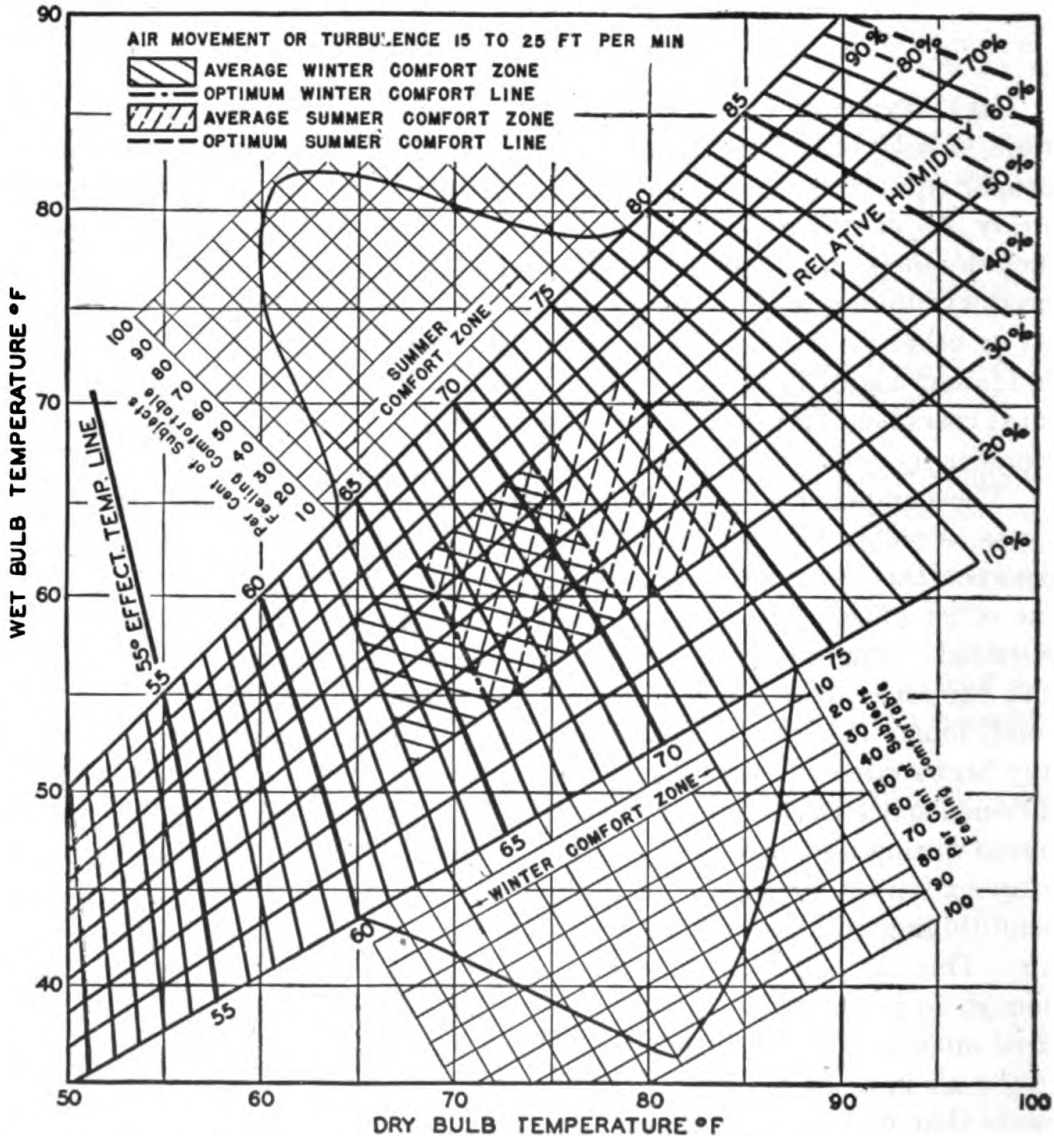


Fig. 10.1. ASHVE comfort chart for still air. Copyright 1948. From *Heating Ventilating Air Conditioning Guide 1948*, Chapter 12.

saturated air at that numerical temperature and with an air movement of 15 to 25 fpm (still air). Effective temperature, however, is not a true comfort index because discomfort may be experienced at extremely high or low humidities, and no account is taken of the radiation effect of surrounding surfaces. Nevertheless, it is the best comfort index at present for still-air conditions between 30 and 70 per cent relative humidity. It is used when inside design conditions are changed in order to obtain better equipment balance and still produce the same degree of comfort. Figure

10.1 is the ASHVE Comfort Chart, and values in Table 10.1 are taken from it.

Contrary to conclusions from the early tests, recent investigations in various parts of the United States indicate that, for office and other sedentary workers considered as a group, optimum comfort is obtained when the temperature is between 74 and 78 F and that; as long as the relative

TABLE 10.1
APPROXIMATE CONDITIONS OF EQUAL EFFECTIVE TEMPERATURE¹

Effective Temperature	Dry Bulb, F	Wet Bulb, F	Relative Humidity
71°	79	59	29%
	78	60.5	35
	77	62	42
	76	63.5	51
	75	65.5	60
	74	67	69
72	80	61	32
	79	62	38
	78	64	45
	77	65	52
	76	66.5	61
	75	68	70
73	81	62	34
	80	64	41
	79	65.5	48
	78	66.5	55
	77	68	62
	76	69	70
74	83	63	32
	82	64.5	38
	81	66	44
	80	67	51
	79	68	58
	78	69.5	65
75	84	64.5	33
	83	66	39
	82	67	46
	81	68	53
	80	69.5	60
	79	70.5	66
75.5	85	65	33
	84	66	38
	83	67	43
	82	68.5	49
	81	69.5	56
	80	71	64

¹ Based on data from Fig. 10 of Chapter 12, *Heating Ventilating Air Conditioning Guide 1948*. New York: American Society of Heating and Ventilating Engineers.

TABLE 10.2
STORAGE CONDITIONS AND PROPERTIES OF PERISHABLE PRODUCTS²

Product	Range of Storage Temp. F		Optimum Humidity Per Cent		Freezing Point, F	Composition, % Water	Specific Heat, Btu per lb per F		Latent Heat of Fusion, Btu/lb	Maximum Storage Period
	Retail and Wholesale	Ware-house Storage	Gravity Air Circulation	Forced Air Circulation			Above Freezing	Below Freezing		
<i>Fruits</i>										
Apples.....	35-40	30-32	85	88	28.5	85	.90	.49	122	8 months
Apricots.....	35-40	31-33	80	85	29	85	.92	.50	122	12 days
Bananas.....	55-56	55-56	80	80	26-30	75	.90	10 days
Berries.....	35-40	31-33	80	85	26-29	84	.90	.49	120	7-20 days
Cantaloupe.....	50-55	50-55	75	85	30.5	45	.56	.33	66	20 days
Figs and dates.....	40-50	34-36	70	75	28	80	.88	.48	116	15 days
Grapes.....	35-40	30-32	80	85	28	77	.90	.61	112	1-6 months
Grapefruit.....	40-45	32-34	85	90	28.5	77	.87	.49	112	90 days
Lemons.....	50-60	50-55	80	85	28	89	.94	.50	130	90 days
Oranges.....	40-45	32-34	80	85	28	86	.90	.47	124	2 months
Peaches.....	35-40	31-33	80	85	29.5	88	.92	.48	128	30 days
Pears.....	35-40	30-32	85	90	28.5	84	.91	.49	122	1-7 months
Persimmons.....	40-45	33-35	80	85	28.5	66	.80	.46	96	15 days
Pineapples.....	40-45	38-40	75	85	28	88	.90	.50	128	30 days
Plums.....	35-40	31-33	80	85	28.5	84	.91	.49	120	15 days
Pomegranates.....	40-45	33-35	80	85	28	77	.87	.48	112	90 days
Prunes.....	35-40	31-33	80	85	28	80	.88	.48	116	15 days
Strawberries.....	35-40	31-33	80	85	30	90.5	.92	.48	131	10 days
<i>Vegetables</i>										
Artichokes.....	40-45	31-33	90	90	27.5	92	.90	.49	134	2-5 months
Asparagus.....	40-45	32-34	90	90	30	94	.91	.49	136	30 days
Avocados.....	40-45	33-35	85	85	31	94	.91	.49	136	10 days
Beans (string).....	40-45	32-34	85	90	30	68.5	.80	.46	98.5	30 days
Beans (dried).....	50-60	32-35	60	60	12.5	.30	.25	18	12 months
Beets.....	40-45	32-34	85	90	27	88.5	.86	.48	128	7-90 days
Broccoli.....	40-45	32-34	85	90	29	93	.90	.48	135	15 days
Brussels sprouts.....	40-45	32-34	85	95	31	94.5	.91	.49	136	10 days
Cabbage.....	35-40	32-34	90	95	31	91.5	.93	.47	132	4 months
Carrots.....	35-40	32-34	90	95	29.5	88	.86	.45	126	2-4 months
Casabas.....	50-60	50-55	75	85	30	91.5	.90	.46	132	40 days
Cauliflower.....	35-40	32-34	85	90	30	92.5	.90	.46	133	10 days
Celery.....	35-40	31-33	90	90	30	94.5	.91	.46	136	2-4 months

TABLE 10.2 (Continued)
STORAGE CONDITIONS AND PROPERTIES OF PERISHABLE PRODUCTS¹

Product	Range of Storage Temp. F		Optimum Humidity Per Cent		Freezing Point, F	Composition, % Water	Specific Heat, Btu per lb per F		Latent Heat of Fusion, Btu/lb	Maximum Storage Period
	Retail and Wholesale	Ware-house Storage	Gravity Air Circulation	Forced Air Circulation			Above Freezing	Below Freezing		
Miscellaneous										
Beer (wood kegs).....	35-40	34-38	85	85	28	92	1.0	6 months
Beer (steel kegs).....	35-40	34-38	70	70	28	92	1.0	6 months
Butter.....	45-50	80	80	75-70	15	.64	10 days
Butter.....	35-40	15	80	80	30-0	15	.64	.34	15	6 months
Caviar (tub).....	40-45	34-36	85	85	20	15 days
Cheese (curing).....	50-60	50-60	80	85
Cheese (American).....	40-45	32-34	80	80	17	55	.64	.36	79	15 months
Cheese (Camembert).....	40-45	40-45	85	85	18	60	.70	.40	86	90 days
Cheese (Limburger).....	40-45	31-33	85	85	19	60	.70	.40	86	60 days
Cheese (Roquefort).....	40-45	40-45	80	80	3	55	.65	.32	79	60 days
Cheese (Swiss).....	40-45	38-42	80	80	15	55	.64	.36	79	60 days
Chocolate (coating).....	65-70	60-75	55	55	95-85	50	.30	.56	40	6 months
Cream (40%).....	35-40	5-10	80	80	28	55	.85	.40	90	4 months
Eggs (crated).....	40-45	30-31	85	85	27	73	.76	.40	100	12 months
Eggs (frozen).....	15-20	0-5	60	60	2741	100	18 months
Flowers (cut).....	40	35	90	32	1 week
Furs & woolens.....	35	35	65	April to Oct.
Furs & woolens.....	15-50	15-50	70	50	April to Oct.
Honey.....	40-45	31-33	70	70	18	.35	.26	26	12 months
Hops.....	35-40	32-36	65	65	May to Oct.
Ice cream.....	0-10	-20-0	85	85	27-0	60	.78	.45	96	2 weeks
Lard.....	45-50	32-34	80	8052	6 months
Malt.....	50-55	48-52	85	80	May to Oct.
Maple sugar.....	45-50	31-32	70	70	5	.24	.21	7	May to Oct.
Maple syrup.....	45-50	31-32	70	70	36	.49	.31	52	May to Oct.
Milk.....	35-40	35-40	70	70	31	87.5	.93	.49	124	5 days
Nuts (dried).....	35-40	30-32	75	75	3-10	.21-.29	.19-.24	4.3-14	8-12 months
Olivo.....	45-50	34-36	80	8048	90 days
Tobacco and cigars.....	42-44	42-44	90	25
Vaccine serum.....	43-45	43-55	70	70	4 months

¹ From Carrier Document 2D-58 of Dec. 15, 1936. Carrier Corporation, Syracuse, N. Y.

humidity is between 35 and 60 per cent, its actual value is not important. These conditions meet both summer and winter requirements for extended occupancy, and in many establishments the thermostat is kept at the same setting the year around. The above, of course, presumes the absence of drafts.

The actual inside design temperatures selected in comfort air conditioning, however, are not necessarily those of optimum comfort. The length and type of occupancy, the outside design conditions used, and economic considerations affect the choice. When the design outside dry bulb is 95 F or higher, inside design conditions for most commercial air-conditioning applications are 80 F and 50 per cent relative humidity, or slightly under 74 F effective temperature. Conditions up to 75 F effective temperature are sometimes used for drugstores, cigar stores, quick-lunch counters, and similar establishments. For homes, offices, and stores having extended occupancy, an effective temperature of 73 F, or sometimes 72 F, is selected. One degree lower design effective temperature should be used for each 5-degree drop in outside design temperature. The room thermostat is usually set lower than the design temperature.

The inside design conditions for industrial air conditioning and for all refrigerated spaces depend upon the usage of the space and the specific products involved. Not only the kind of product but its age, condition, proposed length of storage, and the processes involved determine the desired temperature, humidity, and air movement. Very few products can be treated alike. Recommended conditions for several industrial air-conditioning processes are given in Chapter 18 under the specific applications. Frequently, the production foreman or superintendent knows under what conditions he can obtain the most satisfactory product, and the air-conditioning or refrigerating system is designed accordingly. In other cases it is the responsibility of the air-conditioning or refrigeration engineer to investigate or experiment and to recommend design conditions. Recommended storage conditions and other information for many food products are given in Table 10.2.

One point to be emphasized is that guesswork should be eliminated in food preservation. There is no need for assuming design conditions with the great volume of information on the storage and freezing of most food products available from the U.S. Department of Agriculture, the various state agricultural departments and colleges, commercial concerns, recent trade magazines, and the ASRE Data Books. It is urged that before designing, recommending, or buying a system or equipment in this field the latest reliable information available be studied.

Outside design conditions are determined from local experience or codes. They usually are the average maximum values, excluding the high values that occurred for less than three hours in only 10 days out of the year. Table 10.3 lists recommended values.

TABLE 10.3
OUTSIDE DESIGN CONDITIONS³

State	City	Winter (Heating) Dry Bulb, F	Summer (Cooling)	
			Dry Bulb, F	Wet Bulb, F
Alabama	Birmingham	10	95	78
	Mobile	15	95	80
	Montgomery	10	95	78
Arizona	Flagstaff	-10	90	65
	Phoenix	25	105	76
	Tucson	25	105	72
	Yuma	30	110	78
Arkansas	Little Rock	5	95	78
California	Bakersfield	25	105	70
	El Centro	25	110	78
	Fresno	25	105	70
	Long Beach	35	90	70
	Los Angeles	35	90	70
	Needles	25	115	80
	Oakland	30	85	65
	Pasadena	30	95	70
	Sacramento	30	100	72
	San Bernadino	30	105	72
	San Diego	35	85	68
	San Francisco	35	85	65
Colorado	Denver	-10	95	64
Connecticut	Bridgeport	0	95	75
	Hartford	0	93	75
	New Haven	0	95	75
Delaware	Wilmington	0	95	78
District of Columbia	Washington	0	95	78
Florida	Jacksonville	25	95	78
	Miami	35	91	79
	Pensacola	20	95	78
	Tampa	30	95	78
Georgia	Atlanta	10	95	76
	Augusta	10	98	76
	Brunswick	20	95	78
	Columbus	10	98	76
	Savannah	20	95	78
Idaho	Boise	-10	95	65
Illinois	Chicago	-10	95	75
	Peoria	-10	95	76

TABLE 10.3 (Continued)
OUTSIDE DESIGN CONDITIONS³

State	City	Winter (Heating) Dry Bulb, F	Summer (Cooling)	
			Dry Bulb, F	Wet Bulb, F
Indiana	Fort Wayne	-10	95	75
	Indianapolis	-10	95	76
Iowa	Des Moines	-15	95	77
	Sioux City	-20	95	77
Kansas	Wichita	-10	100	75
Kentucky	Louisville	0	95	78
Louisiana	New Orleans	20	95	78
	Shreveport	10	100	78
Maine	Augusta	-15	90	73
	Bangor	-20	90	73
	Portland	- 5	90	73
Maryland	Baltimore	0	95	78
	Cumberland	0	95	75
Massachusetts	Boston	0	92	75
	Fitchburg	-10	93	75
	Springfield	-10	93	75
	Worcester	- 5	93	75
Michigan	Detroit	-10	95	75
	Flint	-10	95	75
	Grand Rapids	-10	95	75
	Saginaw	-10	95	75
Minnesota	Duluth	-30	93	73
	Minneapolis	-20	95	75
	St. Paul	-20	95	75
Mississippi	Vicksburg	10	95	78
Missouri	Kansas City	-10	100	75
	St. Louis	-10	95	78
Montana	Billings	-25	90	66
	Helena	-20	95	65
	Missoula	-15	95	66
Nebraska	Lincoln	-15	95	77
	Omaha	-15	95	77
Nevada	Reno	- 5	95	77

TABLE 10.3 (Continued)
OUTSIDE DESIGN CONDITIONS³

State	City	Winter (Heating) Dry Bulb, F	Summer (Cooling)	
			Dry Bulb, F	Wet Bulb, F
New Hampshire	Concord	-15	90	73
	Manchester	-10	90	73
	Portsmouth	0	90	73
New Jersey	Jersey City	0	95	75
	Newark	0	95	75
	Trenton	0	95	78
New Mexico	Santa Fe	- 5	95	65
New York	Albany	-10	93	75
	Buffalo	- 5	93	75
	New York	0	95	75
	Rochester	- 5	93	75
	Syracuse	-10	93	75
North Carolina	Asheville	0	93	75
	Charlotte	5	95	75
	Greensboro	5	95	75
	Raleigh	5	95	76
	Wilmington	15	95	78
North Dakota	Bismarck	-30	95	73
Ohio	Akron	- 5	95	75
	Cincinnati	0	95	78
	Cleveland	- 5	95	75
	Columbus	-10	95	76
	Dayton	0	95	76
	Toledo	-10	95	75
	Youngstown	- 5	95	75
Oklahoma	Oklahoma City	0	101	77
	Tulsa	0	101	77
Oregon	Baker	- 5	90	66
	Portland	10	90	68
	Roseburg	10	90	66
Pennsylvania	Altoona	- 5	95	75
	Erie	- 5	93	75
	Harrisburg	0	95	75
	Oil City	-15	95	75
	Philadelphia	0	95	78
	Pittsburgh	- 5	95	75
	Scranton	- 5	95	75
Rhode Island	Pawtucket	0	93	75
	Providence	0	93	75

TABLE 10.3 (Continued)
OUTSIDE DESIGN CONDITIONS³

State	City	Winter (Heating) Dry Bulb, F	Summer (Cooling)	
			Dry Bulb, F	Wet Bulb, F
South Carolina	Charleston	15	95	78
	Columbia	10	95	75
	Greenville	10	95	75
South Dakota	Sioux Falls	-20	95	75
Tennessee	Chattanooga	10	95	76
	Knoxville	0	95	75
	Memphis	0	95	78
	Nashville	0	95	78
Texas	Dallas	10	100	78
	El Paso	10	100	69
	Fort Worth	10	100	78
	Galveston	20	95	80
	Houston	20	95	78
	San Antonio	20	95	78
Utah	Salt Lake City	-10	95	65
Vermont	Burlington	-15	90	73
	Rutland	-15	90	73
Virginia	Norfolk	15	95	78
	Richmond	10	95	78
	Roanoke	0	95	76
Washington	Seattle	15	85	65
	Spokane	-15	93	65
	Tacoma	15	85	64
	Walla Walla	-5	95	65
	Wenatchee	-10	90	65
	Yakima	-5	95	65
West Virginia	Bluefield	-10	95	75
	Charleston	0	95	75
	Huntington	-5	95	76
	Parkersburg	-10	95	75
	Wheeling	-5	95	75
Wisconsin	Madison	-15	95	75
	Milwaukee	-15	95	75
Wyoming	Cheyenne	-15	95	65

³ Reprinted, by permission, from *Application Engineering Standards for Air Conditioning for Comfort*, published by Air Conditioning and Refrigerating Machinery Association, Inc., 1947.

In some applications the maximum cooling load does not occur in the afternoon, when the outdoor temperature is usually at the maximum. Probable variations in outside conditions for typical design days of 95 F dry bulb and 78 F and 75 F wet bulbs are shown in Table 10.4.

TABLE 10.4
PROBABLE OUTSIDE TEMPERATURE VARIATIONS FOR TWO TYPICAL DESIGN DAYS
WITH CONSTANT DEW POINT*

Standard Time	Dry Bulb, F	Wet Bulb, F	
10 A.M.	88	73	76
12 noon	92	74	77
2 P.M.	95	75	78
4 P.M.	95	75	78
6 P.M.	94	75	78
8 P.M.	91	74	77
10 P.M.	88	73	76

* (1) Dry-bulb temperature variations based on data for a typical July day with a mean daily range of 15 F as presented by Faust *et al.*, "A Rational Heat Gain Method for the Determination of Air Conditioning Cooling Loads," *Transactions ASHVE*, Vol. 41 (1935), pp. 331-332.

(2) Wet-bulb temperature variations based on approximate constant daily dew-point temperature.

(3) Also see The Marley Co., "Summer Weather Data"; and Carrier, Cherne, and Grant, *Modern Heating, Ventilating and Air Conditioning*. New York: Pitman Publishing Corporation, 1940, p. 36.

10.3. Air-Conditioning-Load Items. In general, the components of the cooling loads for comfort air conditioning, for industrial air conditioning, or for any refrigerated space are the same. These may be classified in accordance with the following items, but all items are not always present in every case.

A. Sensible heat gains in the space:

1. Heat transmission through the structure
2. Solar radiation
3. Infiltration or air leakage into the space
4. Heat emission from occupants
5. Heat from electric lights
6. Heat to be extracted from materials or products brought in at higher than room temperature
7. Heat from other internal sources, such as motors and chemical, mechanical, gas, steam, hot water, electrical, or other appliances present

B. Latent heat gains in the space:

1. Infiltration by air leakage and by vapor-pressure difference
2. Moisture from occupants

3. Moisture from materials or products in the space
4. Moisture from other internal sources such as wet surfaces and chemical, gas, steam, hot water, electrical or other appliances

C. Outside ventilation air:

1. Sensible heat gain due to temperature difference
2. Latent heat gain due to moisture difference

D. Miscellaneous items

10.4. Building Heat Transmission. After the proper design conditions are selected, the next step is to calculate the sensible heat load. The first item is heat gain by transmission through the surrounding walls, windows, doors, floor, and ceiling. As was pointed out in Chapter 8, when the temperature on one side of a wall differs from that on the other, heat flows from the high-temperature side to the low-temperature side. If the temperature is the same on both sides, there is no heat transfer. For example, on a cold winter day heat flows from a heated room to the outside, and on a hot summer day heat flows from outside into an air-cooled room. If the adjoining space is cooled to the same temperature as that in the space under consideration, there is no heat transmission between the spaces. The method of calculating the summer heat gain is similar to that for calculating the winter heat loss. The heat transferred under steady heat flow through a given part of the structure is equal to the over-all heat-transmission coefficient for that portion times the total internal surface area involved times the difference between inside and outside air temperatures. In equation form:

$$q = UA\Delta t \text{ Btu per hour} \quad (8.26)$$

The value of U may be determined from test data or calculated from the following equation as explained in §8.16:

$$U = \frac{1}{\frac{1}{f_o} + \frac{1}{C} + \frac{x_1}{k_1} + \frac{1}{a} + \frac{x_2}{k_2} + \frac{1}{f_i}} \text{ Btu per (hr)(sq ft)(F)} \quad (8.27)$$

Values to use in equations 8.26 and 8.27 are given in Table 10.5 and in the Appendix, Tables A.14, A.15, and A.16. Additional values may be found in bibliographic references 3 and 4 at the end of the chapter.

The temperature difference $\Delta t = t_o - t_i$.

The area A in equation 8.26 may be obtained easily by measurement or from drawings, and the temperature difference is obvious when t_o , the temperature in the adjoining space, is constant. For interior walls or partitions, floors, and ceilings, the temperature in the adjoining unconditioned space is usually considered to be 5 to 10 deg below the outdoor

design temperature. Of course, kitchens, boiler rooms, attics, and show windows are exceptions, and the temperature in any one of them may often be higher than the outdoor temperature.

In the case of exterior walls or ceilings, that is, walls exposed to the outdoors or ceilings directly under a roof, variations in outdoor temperatures and in the solar effect produce unsteady or periodic heat flow. The

TABLE 10.5
TRANSMITTANCE U , FOR STRUCTURES⁴

In Btu per (hr)(sq ft)(F) (difference in temperature between the air on the two sides) and with a wind velocity of 15 miles per hour outside

Structure	U
Brick wall, 8-in., bare.....	0.50
Brick wall, 8-in., plaster one side on brick.....	0.46
Brick wall, 8-in., plaster one side on metal lath-furred.....	0.32
Brick wall, 16-in., bare.....	0.28
Brick wall, 16-in., plaster one side on brick.....	0.27
Brick wall, 16-in., plaster one side on metal lath-furred.....	0.21
Hollow tile, stucco exterior, 8-in., bare.....	0.40
Hollow tile, stucco exterior, 8-in., plaster on metal lath-furred...	0.28
Hollow tile, stucco exterior, 12-in., bare.....	0.30
Hollow tile, stucco exterior, 12-in., plaster on metal lath-furred..	0.22
Cinder blocks, 8-in., bare.....	0.42
Cinder blocks, 8-in., plaster one side on metal lath-furred.....	0.28
Concrete blocks, 8-in., bare.....	0.56
Clapboard frame construction, plaster on wood lath.....	0.25
Wood shingle frame construction, plaster on wood lath.....	0.25
Stucco frame construction, plaster on wood lath.....	0.30
Brick veneer frame construction, plaster on wood lath.....	0.27

time lag of heat transmission that depends upon the type of wall or ceiling structure must be considered.

In the past most engineers neglected the time factor for continually shaded walls or roofs and used t_o as 0 to 12 F less than the outside design temperature, depending upon the mass of the wall or roof. They did consider the time lag to a certain degree for sun-exposed surfaces and used a t_o from 5 F below to 60 F above the outdoor design temperature, depending on the time of day, orientation, and type of construction. More reliable data based upon tests conducted in comparatively recent years are now available. This information has been summarized in sets of curves showing heat flow and time relationships for roofs and walls of various constructions and exposures. Copies of these curves are shown in Figs. 10.2 to 10.7. When they are used the following equation is applied:

$$q = AH_T \text{ Btu per hour} \quad (10.1)$$

where A = inside surface area in sq ft

H_T = heat flow entering the space in Btu per (hr)(sq ft) taken from the proper curve

⁴ From *Refrigerating Data Book*, 5th ed., 1943. New York: American Society of Refrigerating Engineers, p. 157.

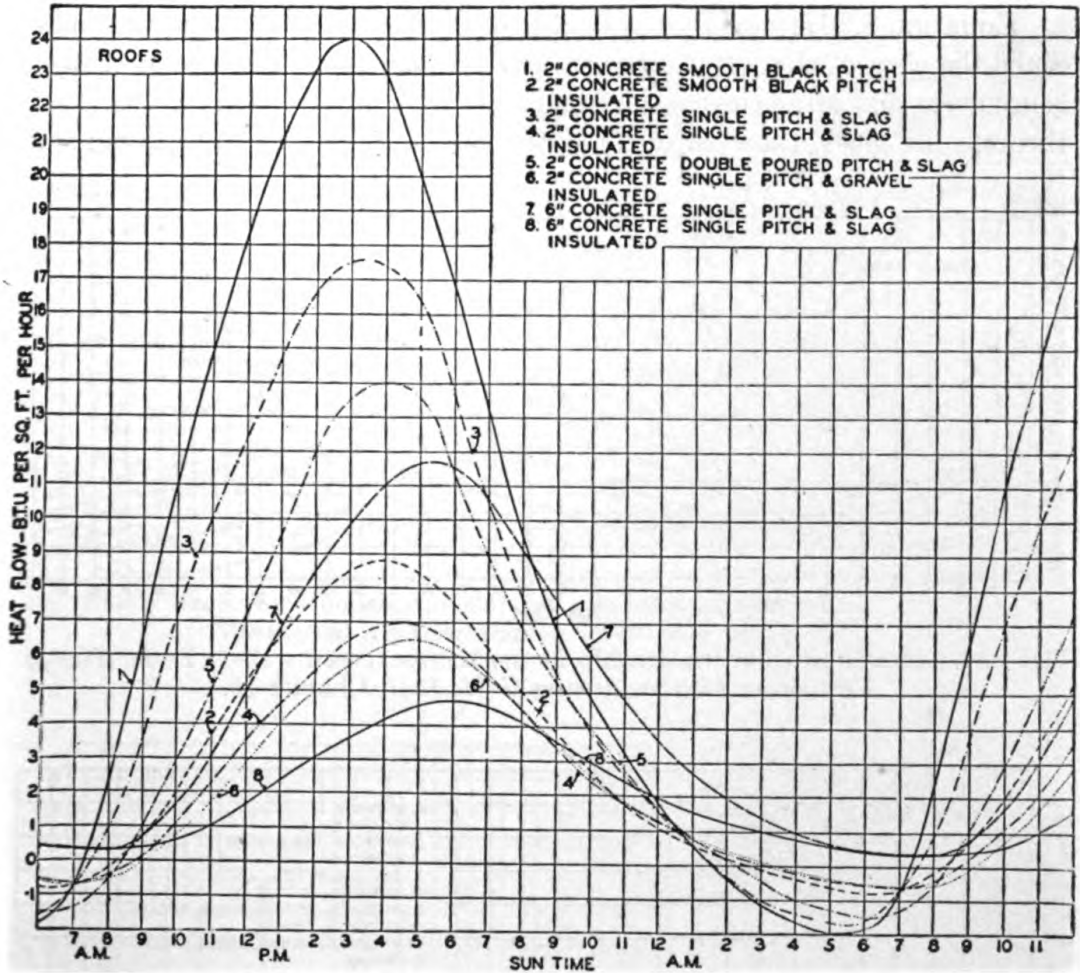


Fig. 10.2. Heat flow-time relationship for horizontal roofs. From *Heating Ventilating Air Conditioning Guide 1947*, Chapter 15.

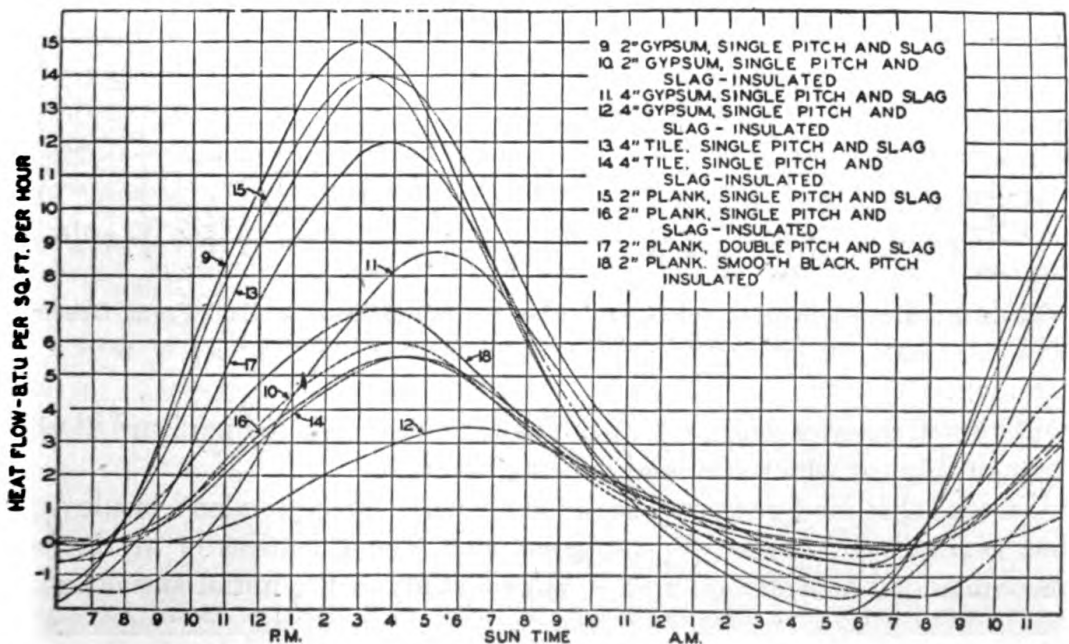


Fig. 10.3. Heat flow-time relationship for horizontal roofs. From *Heating Ventilating Air Conditioning Guide 1947*, Chapter 15.

Equation 8.26 is used for the heat transfer through doors and windows. Since the area of the wood or metal portion of the ordinary sash or door is not great and often the conduction is not appreciably different than that through the glass, the area of the entire opening is used. For doors and

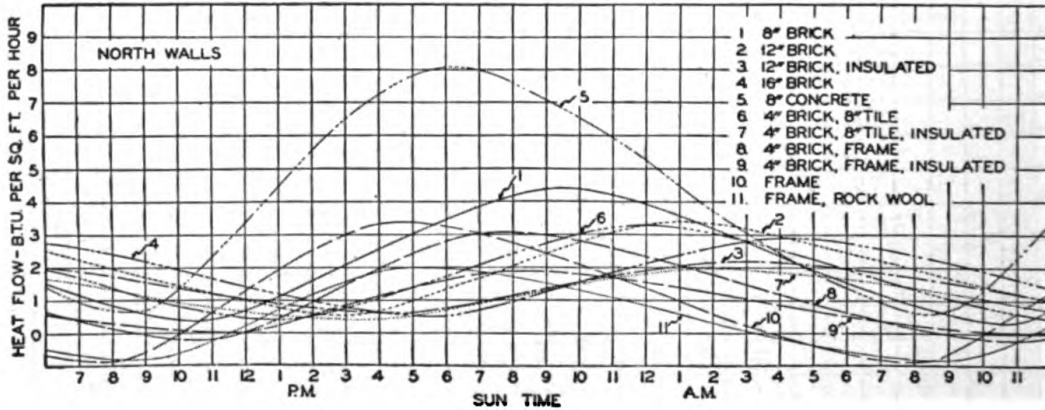


Fig. 10.4. Heat flow-time relationship for northern-exposed walls. From *Heating Ventilating Air Conditioning Guide 1947*, Chapter 15.

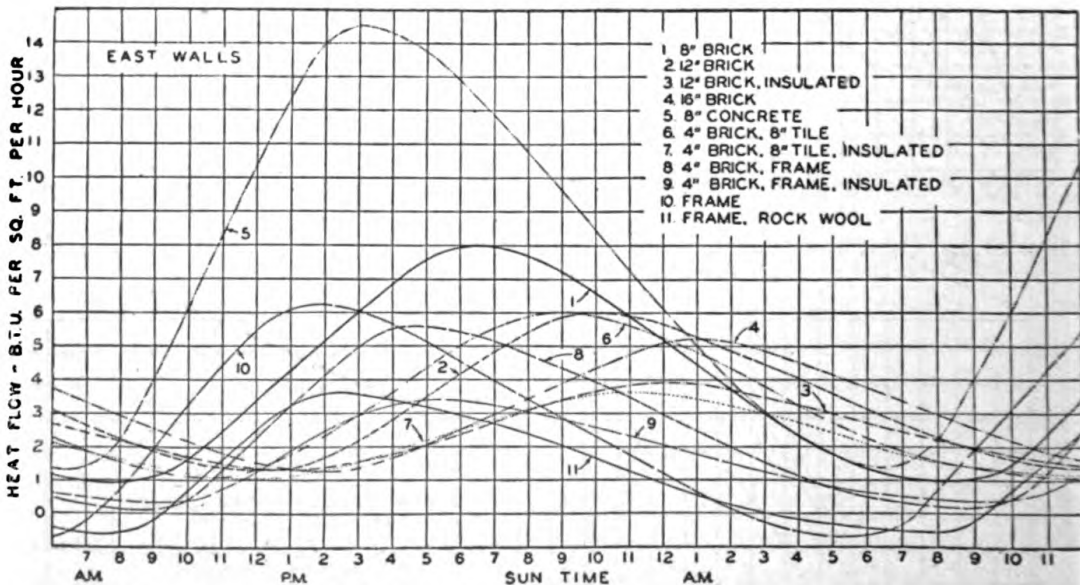


Fig. 10.5. Heat flow-time relationship for eastern-exposed walls. From *Heating Ventilating Air Conditioning Guide 1947*, Chapter 15.

windows in exterior walls, t_o is taken as the outside-air temperature at the time of day for which the load is being estimated.

Cork, shredded redwood bark, glass wool, or equivalent insulation has generally been used for refrigerators. The thickness of insulation recommended depends upon an economic study of the initial and operat-

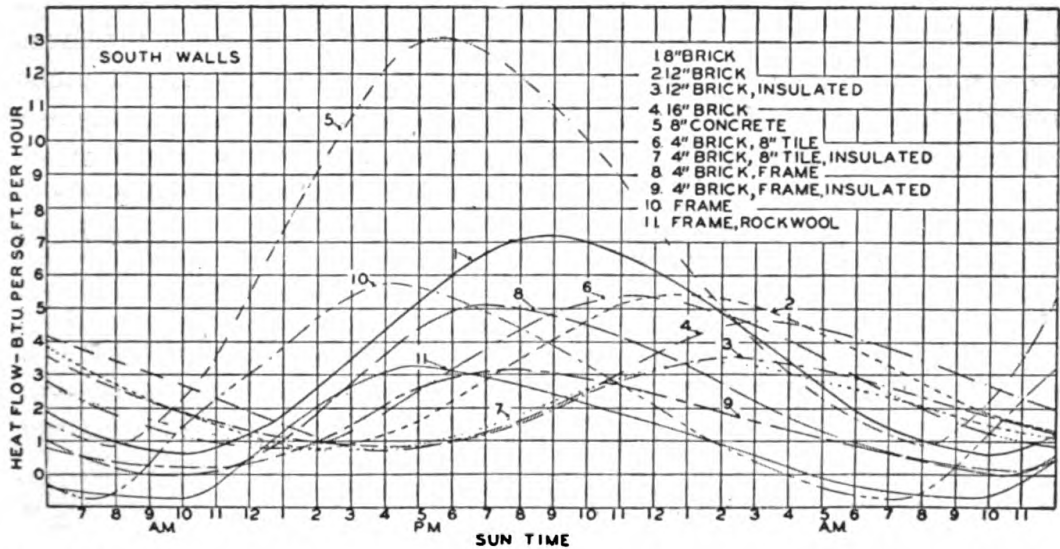


Fig. 10.6. Heat flow-time relationship for southern-exposed walls. From *Heating Ventilating Air Conditioning Guide 1947*, Chapter 15.

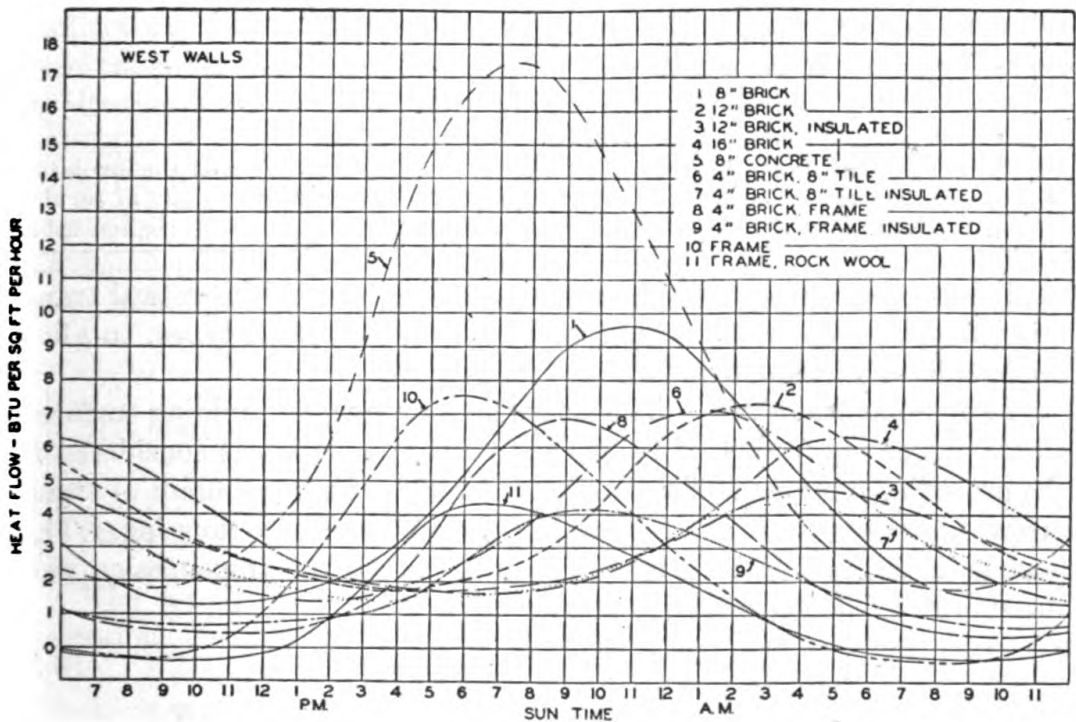


Fig. 10.7. Heat flow-time relationship for western-exposed walls. From *Heating Ventilating Air Conditioning Guide 1947*, Chapter 15.

ing costs of the refrigerator and equipment involved. In general, the thicknesses shown in Table 10.6 are recommended. For walls with sun exposure, an additional inch is suggested, with two additional inches recommended for roofs. For walk-in refrigerators, this insulation is used in addition to a masonry or other substantial wall, and $\frac{1}{2}$ -in. Portland cement is usually applied to the inside surface. A good vapor barrier should also be used. Vapor-tight double metal containers are now used for reach-in refrigerators and freezer units.

TABLE 10.6
MINIMUM THICKNESS OF INSULATION⁵

Room Temperature, F	Equivalent Thickness of Corkboard, in.*	Minimum Transmission Coefficient Btu per (hr) (sq ft)(F)†
-19 to -10	8	.042
- 9 to - 5	7	.047
- 4 to + 5	6	.055
6 to 20	5	.067
21 to 35	4	.083
36 to 45	3	.111
46 and over	2	.167

* These recommended thicknesses are based on the assumption that the insulation is properly applied and remains in a reasonably dry condition over years of service.

† Units have been converted from Btu per 24-hr period as given in original table.

It is recommended that 15 per cent be added to the theoretical transmission values when computing refrigerator loads of all types, to allow for installation irregularities.

10.5. Solar Radiation. The solar effect on opaque building surfaces, where a time lag from 1 to 16 hr or more is involved, was considered in the preceding section. The solar heat gain through transparent or translucent materials, such as windows, involves little if any time lag. This heat gain through glass is figured in addition to the transmission gain through it. The general equation used is

$$q = A_g f I \quad \text{Btu per hour} \quad (10.2)$$

where A_g = actual area of the glass only, which is usually 80 per cent or less of the opening area used for the transmission gain.
(See §10.4.)

⁵ Adapted, by permission, from *Equipment Standards* of the Air Conditioning and Refrigerating Machinery Association, Inc., 1946 ed.

I = intensity of solar radiation striking the surface in Btu per (hr)(sq ft) and depends upon time of day, orientation, and latitude

f = per cent transmitted to the inside expressed as a decimal and depends on the type of shading appurtenance used

Values of I and f are included in Tables 10.7 and 10.8.

Solar radiation is often the largest component of the room sensible-heat load for a building with considerable window area. It may be necessary to calculate loads for different hours of the day in order to find when and what the maximum load may be. Dividing the building into several zones separately controlled is recommended to overcome the variation in solar radiation and in each zone's load as the day passes.

10.6. Infiltration. This item is estimated in the same manner as it is for heating, but for large spaces it is not proportionately so large an item in the total load. When the normal infiltration does not provide adequate ventilation, outside air is drawn directly into the air-conditioning unit after mixing with some return air and then discharged into the conditioned space. Often the outside air required is sufficient to build up a slight pressure within the space and offset the infiltration. No infiltration need be figured if the volume of outside air handled by the equipment is great enough to offset the larger of the total calculated infiltration or the quantity of air being exhausted, if any. If the outside air handled is not sufficient to do this, the additional requirement is included in the load as infiltration items. Frequent door openings or doors left open cause considerable infiltration. Recommended values in air-conditioning applications are given in Table 10.9. The sensible-heat gain from infiltration is calculated from the equation

$$q = cfm \times 60 \times \rho \times c_p(t_o - t_i) \text{ Btu per hour} \quad (10.3)$$

If ρ is taken as 0.075 (standard air),

$$q = cfm \times 1.08 \times (t_o - t_i) \text{ Btu per hour} \quad (10.4)$$

At 32 F,

$$q = cfm \times 1.16 \times (t_o - t_i) \quad (10.5)$$

The latent heat gain from air infiltration is calculated from the equation

$$q = cfm \times 60 \times \rho \times (w_{H_o} - w_{H_i}) \times \frac{h_{fg}}{7000} \text{ Btu per hour} \quad (10.6)$$

Again taking ρ as 0.075 and h_{fg} as 1060 for 60 F,

$$q = cfm \times 0.68 \times (w_{H_o} - w_{H_i}) \text{ Btu per hour} \quad (10.7)$$

At 32 F,
$$q = cfm \times 0.74 \times (w_{H_o} - w_{H_i}) \text{ Btu per hour} \quad (10.8)$$

TABLE 10.7

TOTAL INSTANTANEOUS RATES OF HEAT GAIN THROUGH SINGLE, UNSHADED, COMMON GLASS, FOR VARIOUSLY ORIENTED VERTICAL AND FOR HORIZONTAL POSITIONS^a
Data for solar declination of 17.5 deg—August 1

Sun Time	Solar Altitude β Deg	Total Instantaneous Rate of Heat Gain, Btu Per Hour for Each Square Foot of Unshaded Glass								
		N	NE	E	SE	S	SW	W	NW	Horizontal

25 Deg North Latitude

6 A.M.	7.5	19	77	84	40	3	3	3	3	12
7	20.5	26	146	173	98	10	10	10	10	68
8	34.0	18	148	193	121	13	13	13	13	143
9	47.5	15	118	172	120	16	15	15	15	205
10	61.5	16	67	124	95	22	16	16	16	252
11	74.5	16	25	57	57	25	16	16	16	282
12	83.0	16	16	16	25	28	25	16	16	293
1 P.M.	74.5	16	16	16	16	25	57	57	25	282
2	61.5	16	16	16	16	22	95	124	67	252
3	47.5	15	15	15	15	16	120	172	118	205
4	34.0	18	13	13	13	13	121	193	148	143
5	20.5	26	10	10	10	10	98	173	146	68
6	7.5	19	3	3	3	3	40	84	77	12

30 Deg North Latitude

6 A.M.	9.0	22	88	97	47	4	4	4	4	15
7	21.5	23	146	176	105	11	11	11	11	74
8	34.5	16	140	194	130	14	14	14	14	144
9	47.5	15	104	171	133	20	15	15	15	205
10	60.0	16	53	126	112	33	16	16	16	248
11	72.0	16	19	56	74	42	16	16	16	277
12	78.0	16	16	16	34	45	34	16	16	288
1 P.M.	72.0	16	16	16	16	42	74	56	19	277
2	60.0	16	16	16	16	33	112	126	53	248
3	47.5	15	15	15	15	20	133	171	104	205
4	34.5	16	14	14	14	14	130	194	140	144
5	21.5	23	11	11	11	11	105	176	146	74
6	9.0	22	4	4	4	4	47	97	88	15

35 Deg North Latitude

6 A.M.	10.0	21	97	109	53	4	4	4	4	19
7	22.5	19	143	179	110	11	11	11	11	79
8	34.5	14	133	194	140	15	14	14	14	144
9	46.5	15	90	170	144	28	15	15	15	200
10	58.5	16	38	126	128	47	16	16	16	243
11	68.5	16	16	56	91	62	17	16	16	269
12	73.0	16	16	16	45	68	45	16	16	279
1 P.M.	68.5	16	16	16	17	62	91	56	16	269
2	58.5	16	16	16	16	47	128	126	38	243
3	46.5	15	15	15	15	28	144	170	90	200
4	34.5	14	14	14	14	15	140	194	133	144
5	22.5	19	11	11	11	11	110	179	143	79
6	10.0	21	4	4	4	4	53	109	97	19

TABLE 10.7 (Continued)

TOTAL INSTANTANEOUS RATES OF HEAT GAIN THROUGH SINGLE, UNSHADED, COMMON GLASS, FOR VARIOUSLY ORIENTED VERTICAL AND FOR HORIZONTAL POSITIONS^a

Data for solar declination of 17.5 deg—August 1

Sun Time	Solar Altitude β Deg	Total Instantaneous Rate of Heat Gain, Btu Per Hour for Each Square Foot of Unshaded Glass								
		N	NE	E	SE	S	SW	W	NW	Horizontal
40 Deg North Latitude										
5 A.M.	1.5	7	18	17	6	1	1	1	1	2
6	11.5	23	106	120	62	5	5	5	5	24
7	23.0	15	141	181	118	11	11	11	11	82
8	34.5	14	122	194	147	19	14	14	14	145
9	45.5	15	76	172	156	42	15	15	15	196
10	56.0	16	30	125	144	66	16	16	16	235
11	64.5	16	16	53	110	85	22	16	16	261
12	68.0	16	16	16	62	94	62	16	16	269
1 P.M.	64.5	16	16	16	22	85	110	52	16	261
2	56.0	16	16	16	16	66	144	125	30	235
3	45.5	15	15	15	15	42	156	172	76	196
4	34.5	14	14	14	14	19	147	194	122	145
5	23.0	15	11	11	11	11	118	181	141	82
6	11.5	23	5	5	5	5	62	120	106	24
7	1.5	7	1	1	1	1	6	17	18	2
45 Deg North Latitude										
5 A.M.	2.0	9	23	23	8	1	1	1	1	2
6	12.5	22	111	129	68	6	6	6	6	28
7	23.0	13	135	182	121	11	11	11	11	82
8	33.5	14	116	192	153	24	14	14	14	141
9	44.0	15	63	168	166	55	15	15	15	189
10	53.0	16	22	123	154	88	16	16	16	225
11	60.0	16	16	56	127	113	30	16	16	249
12	63.0	16	16	16	76	119	76	16	16	256
1 P.M.	60.0	16	16	16	30	113	127	56	16	249
2	53.0	16	16	16	16	88	154	123	22	225
3	44.0	15	15	15	15	55	166	168	63	189
4	33.5	14	14	14	14	24	153	192	116	141
5	23.0	13	11	11	11	11	121	182	135	82
6	12.5	22	6	6	6	6	68	129	111	28
7	2.0	9	1	1	1	1	8	23	23	2
50 Deg North Latitude										
5 A.M.	4.5	18	48	48	17	2	2	2	2	5
6	13.5	22	119	139	75	6	6	6	6	32
7	23.5	11	131	183	127	11	11	11	11	84
8	33.0	13	103	190	161	30	13	13	13	138
9	42.0	14	51	165	173	69	14	14	14	179
10	50.0	16	19	122	166	109	16	16	16	214
11	56.0	16	16	55	138	133	41	16	16	235
12	58.0	16	16	16	92	140	92	16	16	242
1 P.M.	56.0	16	16	16	41	133	138	55	16	235
2	50.0	16	16	16	16	109	166	122	19	214
3	42.0	14	14	14	14	69	173	165	51	179
4	33.0	13	13	13	13	30	161	190	103	138
5	23.5	11	11	11	11	11	127	183	131	84
6	13.5	22	6	6	6	6	75	139	119	32
7	4.5	18	2	2	2	2	17	48	48	5

^a ASHVE research data. Reprinted, by permission, from *Heating Ventilating Air Conditioning Guide 1948*, Chapter 15.

TABLE 10.8
EFFECT OF SHADING UPON TOTAL RATES OF INSTANTANEOUS HEAT GAIN THROUGH GLASS⁷

Type of Shading	Finish	f Fraction of Gain through Unshaded Window
Outside Shading Screen: metal slats 0.050 in. wide, spaced 0.063 in. apart and set at 17-deg angle with horizontal.....	Dark	0.20-0.35
Canvas Awning.....	Dark	0.25-0.35
Outside Venetian Blind: slats at 45 deg, extended as an awning without sides to cover approximately two-thirds of window.....	Light	0.35-0.50
Inside Roller Shade: fully drawn.....	Aluminum	Approx. 0.45
Outside Venetian Blind: slats at 45 deg, fully covering window.....	Aluminum	Approx. 0.6
Inside Venetian Blind: slats at 45 deg, fully covering window.....	Aluminum	0.65-0.80
Inside Roller Shade: half drawn.....	Buff	Approx. 0.7
Inside Roller Shade: half drawn.....	Dark	0.90-0.95

TABLE 10.9⁸
OUTSIDE AIR INFILTRATION FOR AIR CONDITIONING*

$$\begin{aligned}
 (H) &= \text{Room Height} & (L) &= \text{Length,} \\
 (W) &= \text{Width} & (G) &= \text{Wall Factor} \\
 &\text{Room with one outside wall, } (G) = 1 \\
 &\text{Two outside walls, } (G) = 1.5 \\
 &\text{Three or more outside walls, } (G) = 2
 \end{aligned}$$

$$CFM = \frac{(H) \cdots \times (L) \cdots \times (W) \cdots \times (G) \cdots}{60} = \cdots$$

* For rooms with weatherstripped windows or storm sash, use 50% of this value.

Note: For each person entering or leaving through a door which opens to the outside (or unconditioned space), the additional infiltration for a 36-inch swinging door may be taken as 100 cubic feet and for a 72-inch revolving door, the additional infiltration (cubic feet per person per passage) may be taken as:

Usage	Freely Revolving Door	Door Equipped with Brake
Infrequent	75	60
Average	60	50
Heavy	40	40

These figures are based on the assumption that there is no wind pressure and that swinging doors are in use in one wall only. Any swinging doors in other walls should be kept closed to insure air conditioning in accordance with these recommended standards.

⁷ ASHVE research data. Reprinted, by permission, from *Heating Ventilating Air Conditioning Guide* 1948, Chapter 15.

⁸ Reprinted, by permission, from *Application Engineering Standards for Air Conditioning for Comfort*, published by Air Conditioning and Refrigerating Machinery Association, Inc., 1947.

For industrial applications requiring low humidities, it is important to consider that infiltration of water vapor, in addition to that from air infiltration, takes place through the walls unless a good vapor barrier is properly installed in the wall. Heat-insulated walls are not necessarily vapor barriers. Masonry, fibrous materials, and oil paints are not good vapor barriers. Two coats of glossy enamel or of aluminum paint or asphalt impregnated papers well sealed are much better. Factors for vapor permeability of various materials are given in the literature.⁹

Most refrigerators are practically airtight. The infiltration around the doors, which are gasketed, is negligible, so that there is air leakage only when doors are opened. Consequently, the frequency and total time of door openings must be considered. Since these factors cannot be predicted, values of room-air changes based on past experience are recommended. Air-change values for average door usage showing the variation due to room size and temperature are given in Table 10.10. These values should be increased up to 50 per cent or greater for heavier than average door usage. The smaller the refrigerator, the greater the importance of this load component, and the more care recommended in calculating it, since it becomes a larger percentage of the total load. Of course these air-change values do not apply when outside air is handled by the cooling equipment. The heat gain is determined from the following equation:

$$q = \text{cu ft room volume} \times \text{air changes per hour} \\ \times \text{air density} \times (h_o - h_i) \text{ Btu per hour} \quad (10.9)$$

10.7. Occupants. Man and many other living creatures may be compared in some respects with an automatically controlled stoker-fired furnace. Food consisting mainly of carbon, hydrogen, oxygen, nitrogen, and certain minerals, which likewise are the main elements in coal, is taken in periodically just as coal is fed into the hopper. Air is continually drawn in, and the oxygen combines with the carbon to form carbon dioxide, which is exhaled, and with the hydrogen to form water vapor which is also exhaled. Some water also leaves through the sweat glands and evaporates from the skin surface as a part of the marvelous control system that maintains our body temperature at about 99 F. Heat is evolved

⁹ "H & V's Reference Data—241 & 242." *Heating and Ventilating*, Vol. 40, No. 3 (March, 1943).

Teesdale, L. V., "Comparative Resistance to Vapor Transmission of Various Building Materials." *Heating, Piping and Air Conditioning: ASHVE Journal Section*, Vol. 14, No. 12 (December, 1942), p. 736.

Rowley, F. B., Algren, A. B., and Lund, C. E., "Condensation Within Walls." *Heating, Piping and Air Conditioning: ASHVE Journal Section*, Vol. 10, No. 1 (January, 1938), p. 49.

Building Materials and Structures Reports. Washington, D.C.: United States Department of Commerce, United States Government Printing Office.

Refrigerating Data Book, 5th ed. New York: American Society of Refrigerating Engineers, 1943. p. 154.

TABLE 10.10
AVERAGE AIR CHANGES PER 24 HR FOR STORAGE ROOMS DUE TO DOOR OPENING
AND INFILTRATION¹⁰

Volume cu ft	Air changes per 24 hr	Volume cu ft	Air changes per 24 hr	Volume cu ft	Air changes per 24 hr	Volume cu ft	Air changes per 24 hr
<i>Above 32 F</i>							
250	38.0	1,000	17.5	6,000	6.5	30,000	2.7
300	34.5	1,500	14.0	8,000	5.5	40,000	2.3
400	29.5	3,000	12.0	10,000	4.9	50,000	2.0
500	26.0	3,000	9.5	15,000	3.9	75,000	1.6
600	23.0	4,000	8.2	20,000	3.5	100,000	1.4
800	20.0	5,000	7.2	25,000	3.0		
<i>Below 32 F</i>							
250	29.0	1,000	13.5	5,000	5.6	25,000	2.3
300	26.2	1,500	11.0	6,000	5.0	30,000	2.1
400	22.5	2,000	9.3	8,000	4.3	40,000	1.8
500	20.0	2,500	8.1	10,000	3.8	50,000	1.6
600	18.0	3,000	7.4	15,000	3.0	75,000	1.3
800	15.3	4,000	6.3	20,000	2.6	100,000	1.1

from this oxidation process, termed metabolism, as it is from the oxidation or combustion process in the furnace. Since the body surface temperature is about 90 F, sensible heat is lost by radiation, convection, and conduction to the surroundings that are at a lower temperature. Latent heat is added to the room through the moisture exhaled and evaporated from the skin. The higher the room temperature, the greater is the ratio of latent heat to sensible heat given up.

The total amount of heat lost by a person depends upon his activity and the surrounding temperatures. These quantities for an average man, 5 ft 8 in. tall and weighing 150 lb, are given in Table 10.11. Additional values are given in Table 10.12. The average maximum number of occupants is multiplied by the proper factor for the sensible-heat component and by the proper corresponding factor for the latent-heat component of the room load.

$$q = \text{No. of occupants} \times \text{Btu per hr per occupant} \quad \text{Btu per hr} \quad (10.10)$$

10.8. Electric Lights. The total wattage of lights that will be on during the hour selected for figuring the design load is multiplied by 3.4 to obtain the sensible-heat gain in Btu per hour. Exceptions are encountered where some of the lights are recessed in the ceiling or walls, or where high rooms may permit stratification of heat at the ceiling. In such cases

¹⁰ From *Refrigerating Data Book*, 5th ed., 1943. New York: American Society of Refrigerating Engineers, p. 170.

a reduction factor may be employed. In some spaces the lights near the window are turned off when the sun is shining on the window; in other spaces the lights are left on continuously.

$$q = \text{effective wattage} \times 3.4 \quad \text{Btu per hour} \quad (10.11)$$

10.9. Product Load. This component may consist of several subdivisions, depending upon the product and temperatures involved. These subdivisions, each of which must be calculated separately, are as follows:

Chilling above freezing
Freezing
Cooling below freezing
Product reaction or respiration heat

Five factors must be known in order to determine the product chilling load: (1) The entering-product temperature should be obtained, possibly from the man in charge of the operations in the space. (2) The final product temperature desired should be selected, usually from tables based on recent experiences. (3) The maximum pounds of product that are

TABLE 10.11
HEAT LOSS FROM THE AVERAGE HUMAN BODY¹¹

Room Dry Bulb	Per Person at Rest or Moving Slowly			Per Person Doing Light Work		
	Sensible Heat	Latent Heat	Total Heat	Sensible Heat	Latent Heat	Total Heat
86	160	240	400	110	550	660
84	180	220	400	150	510	660
82	200	200	400	180	480	660
80	220	180	400	210	450	660
78	240	160	400	240	420	660
76	260	140	400	270	390	660
74	280	120	400	300	360	660
72	290	110	400	330	330	660
70	300	100	400	350	310	660
60	460	200	660
50	540	120	660
40	620	110	730
35	650	110	760
30	690	110	800
25	730	110	840

¹¹ Based on data from Figs. 6 and 7 of Chapter 12, *Heating Ventilating Air Conditioning Guide 1948*.

TABLE 10.12
HEAT GAIN FROM OCCUPANTS¹²

Degree of Activity	Typical Application	Total Heat Adult Male Btu/Hr	Total Heat Ad-justed* Btu/Hr	Sensible Heat Btu/Hr	Latent Heat Btu/Hr
Seated at rest	Theater—Matinee	390	330	180	150
	Theater—Evening	390	350	195	155
Seated, very light work	Offices, Hotels, Apartments	450	400	195	205
Moderately active office work	Offices, Hotels, Apartments	475	450	200	250
Standing, light work; walking slowly	Dept. Store, Retail Store, 10¢ Store	550	450	200	250
Walking; seated; standing; walking slowly	Drug Store	550	500	200	300
	Bank				
Sedentary work	Restaurant†	490	550	220	330
Light bench work	Factory	800	750	220	530
Moderate dancing	Dance Hall	900	850	245	605
Walking, 2 mph; moderately heavy work	Factory	1000	1000	300	700
Bowling‡	Bowling Alley	1500	1450	465	985
Heavy work	Factory	1500	1450	465	985

* Adjusted total heat gain is based on normal percentage of men, women, and children for the application listed and is based on the gain from an adult female being 85% of the value for an adult male and the gain from a child being 75% of the value for an adult male.

† The adjusted total heat value for sedentary work, restaurant, includes 60 Btu per hour for food per individual (30 Btu sensible and 30 Btu latent).

‡ For bowling, figure one person per alley actually bowling and all others as sitting (400 Btu per hour) or standing (550 Btu per hour).

Note: The above values are based on 80 F room dry-bulb temperature. For 78 F room dry-bulb temperature, the total heat gain remains the same, but the sensible heat values should be increased by approximately 10% and the latent heat values decreased accordingly.

to be chilled at any one time must be decided. (4) The chilling time required for the kind and individual size of product handled can be obtained from test data available. There is a fixed minimum time for chilling that cannot be reduced appreciably even by lowering the room temperature. In other words, heat, and incidentally moisture, can be removed from a product at a certain maximum rate. It is poor engineering to figure on quick-chilling a large product and to install the extra capacity calculated when such rapid chilling actually is impractical or

¹² Reprinted, by permission, from *Application Engineering Standards for Air Conditioning for Comfort*, published by Air Conditioning and Refrigerating Machinery Association, Inc., 1947.

even impossible. (5) The specific heat of the product can be found from tables.

When figuring a chilling load on an hourly basis, an additional "chilling factor" is recommended to allow for the high load at the start of the chilling period. This factor varies for different products and is equivalent to adding up to 50 per cent more to the average hourly chilling load. The chilling load in equation form is

$$q = \frac{\text{lb} \times \text{sp ht} \times (t_1 - t_2)}{\text{chilling time in hr} \times \text{chilling factor}} \text{ Btu per hr} \quad (10.12)$$

The freezing load, sometimes forgotten by the novice in calculating a freezer load, often is the largest item. The factors involved are the weight of product to be frozen, its latent heat of fusion, and the freezing time. In equation form,

$$q = \frac{\text{lb} \times \text{latent heat}}{\text{freezing time in hr}} \text{ Btu per hr} \quad (10.13)$$

The cooling load below freezing involves the product weight, its specific heat when frozen, the temperature drop desired, and the time allowed. The specific heat is considerably less than for the unfrozen product. Incidentally, the freezing temperature for most food products is about 28 F (see Table 10.2). In equation form,

$$q = \frac{\text{lb} \times \text{sp ht below freezing} \times (t_f - t_i)}{\text{cooling time in hr}} \text{ Btu per hr} \quad (10.14)$$

During the maturing of some food products, even in cold storage, reaction or respiration heat is evolved. The rate of heat production is shown in Table 10.13, and multiplying by the weight of product gives the heat gain.

$$q = \text{lb} \times \text{evolution of heat per (lb)(hr)} \text{ Btu per hr} \quad (10.15)$$

When the product is in containers, the cooling of the containers must also be included. Equation 10.12 may be applied to them.

Equations 10.12, 10.13, 10.14, and 10.15 give total heat values, which may be sufficient in many instances. However, refrigeration applications often are essentially no different than industrial air-conditioning applications. For such cases it is recommended that when possible the total room load be divided into sensible and latent heat components in order to select a satisfactory coil size, refrigerant temperature, and also fan capacity, if a fan is used. The humidity and velocity of the supply-air into the room as well as its temperature are important in spaces where the products are not in sealed packages, and proper equipment size and balance is essential in order to maintain the desired conditions. Unfor-

TABLE 10.13
APPROXIMATE RATE OF EVOLUTION OF HEAT BY CERTAIN FRESH FRUITS AND
VEGETABLES WHEN STORED AT THE TEMPERATURES INDICATED¹³

Commodity	Temperature, F	Heat evolved per ton of fruits or vegetables per 24 hours, British thermal units	Commodity	Temperature, F	Heat evolved per ton of fruits or vegetables per 24 hours, British thermal units
Apples	32	660 to 1,000	Grapes—(Cont.)		
	40	1,110 to 1,760	Emperor	32	350
	60	4,400 to 6,600		43	850
	85	6,600 to 15,400		53	1,810
Bananas:			Ohanez	32	300
Green	54	3,300		43	740
Turning	68	8,360		53	1,570
Ripe	68	9,240	Lemons	32	580
	68	8,360		40	810
Beets	32	2,650		60	2,070
	40	4,060		80	6,200
	60	7,240	Lettuce	32	11,320
Cantaloupes	32	1,320		40	15,990
	40	1,960		60	45,980
	60	8,500	Mushrooms (cultivated)	32	6,160
Carrots	32	2,130		50	22,000
	40	3,470		70	58,000
	60	8,080	Onions (Yellow Globe)	32	660 to 1,100
Celery	32	2,820		50	1,760 to 1,980
	40	4,540		70	3,080 to 4,180
	60	13,520	Oranges	32	690 to 900
Cherries (sour)	32	1,320 to 1,760		40	1,400
	60	11 000 to 13,200		60	5,000
				80	8,000
Grapefruit	32	460	Peaches	32	850 to 1,370
	40	1,070		40	1,440 to 2,030
	60	2,770		60	7,260 to 9,310
	80	4,180		80	17,930 to 22,460
Grapes:	36	660 to 1,100	Pears (Bartlett)	32	660 to 880
Cornichon and	60	2,200 to 2,640		60	8,800 to 13,200
Flame Tokay	80	5,500 to 6,600	Peppers	32	2,720
				40	4,700
Sultanina	32	430		60	8,470
	43	1,050			
	53	1,690			

¹³ From *Refrigerating Data Book*, 5th ed., 1943. New York: American Society of Refrigerating Engineers, p. 208

TABLE 10.13 (Continued)
APPROXIMATE RATE OF EVOLUTION OF HEAT BY CERTAIN FRESH FRUITS AND
VEGETABLES WHEN STORED AT THE TEMPERATURES INDICATED¹³

Commodity	Tem- pera- ture, F	Heat evolved per ton of fruits or vegetables per 24 hours, British thermal units	Commodity	Tem- pera- ture, F	Heat evolved per ton of fruits or vegetables per 24 hours, British thermal units
Potatoes (Irish Cobbler)	32 40 70	440 to 880 1,100 to 1,760 2,200 to 3,520	Sweet corn	32 40 60	5,890 8,190 17,130
Raspberries	36 60	4,400 to 6,600 15,400 to 17,600	Sweet potatoes: Not cured	32 40 60	2,440 3,350 6,300
Strawberries	32 40 60 80	2,730 to 3,800 5,130 to 6,600 15,640 to 19,140 37,220 to 46,440	Cured	32 40 60	1,190 1,710 4,280
String beans	32 40 60	4,740 6,740 22,630	Tomatoes (mature green)	32 40 60	580 1,070 6,230
			Tomatoes (ripe)	32 40 60	1,020 1,260 5,640

unately, lack of published information on the ratio of product latent to total heat often makes it difficult to calculate the load and select the equipment from a scientific viewpoint.

The latent heat load involves product dehydration, an interesting though poorly understood subject. At least one company has made available to their engineers some data on this subject, and fragments of information have been published from time to time. Table 10.14 summarizes data compiled by Fiske.

Since attempts to chill or to freeze products in storage rooms have caused trouble, it is strongly recommended that separate chilling spaces and freezing spaces be provided in addition to the storage spaces. The use of moistureproof containers is also recommended wherever practical. Chilling room loads are often unpredictable, and it is suggested that the equipment selected be flexible in operation and that provision for humidification and for the addition of sensible heat in the space or supply-air system be considered. In many cases it has been necessary to add such equipment. Calculations of loads at the start and at the end of the chilling period will give some indication of the difference required

TABLE 10.14
PRODUCT MOISTURE LOSS IN STORAGE¹⁴

Commodity	Temperature, F	Humidity, Per Cent	Period	Loss, Per Cent
Apples.....	{ 32 37 37 59 59	{ 90 95 75 95 75	6-8 mo 1-50 days	{ 3-4 .03 & 1.1 .08 & 3.5 .06 & 2.5 .18 & 8.2
Beef { 2-lb steak.....		90 & 80	1 day	.9 & 1.8
40-lb roast.....		90 & 80	1 day	.4 & .8
160-lb quarter.....		90 & 80	1 day	.2 & .5
Beef.....	Normal storage	Normal storage	1 mo	3.0
Beets.....	32	98	1 mo	2.5
Brussels sprouts.....	{ 32 32	98 85-90	1 mo 2-3 mo	10.0 20-25
Cabbage { curled.....	31	85	6-7 mo	10-15
red.....	31	85	6-7 mo	6-8
white.....	31	85	6-7 mo	8-10
Carrots.....	32	98	1 mo	1.5
Cauliflower.....	32	98	1 mo	11.0
Celery.....	30	90	6-7 mo	8.0
Cheese.....	60	75	10.0
Cucumbers.....	32	98	1 mo	6.0
Currants.....	30	85	2 mo	6.0
Eggs.....	{ 32 32	99 88	3 mo 3 mo	2.0 7.0
Gooseberries.....	30	85	2 mo	6.0
Lima beans.....	32	98	1 mo	4.0
Onions.....	31	85	6-8 mo	6-8
Peaches.....	30	90	1-2 mo	8-12
Pears.....	32	90	4-6 mo	3-6
Peppers.....	32	98	1 mo	3.0
Plums.....	30	85	2 mo	4-6
Spinach.....	30	90	6-7 mo	4-5
Squash.....	32	98	1 mo	1.0
Strawberries.....	34	90	1 mo	4.0

in equipment operating conditions. The rate of chilling must be selected with care.

When only the total load can be calculated, the equipment selection must be based on a room-to-coil temperature difference selected from experience. Maximum values of temperature difference are given in §11.17 for forced-air coils and for gravity coils.

10.10. Other Internal Heat Sources. Motors and other equipment located in the cooled space and giving off only sensible heat are figured as part of the sensible heat gain. If the actual wattage used can be determined, it is multiplied by 3.4 to obtain the equivalent Btu per hour. Fractional-horsepower motors are not so efficient as larger ones, and when

¹⁴ From Fiske, D. L., "Principles of the Refrigeration Low Side," New Rochelle, New York, 1945.

they are fully loaded, the heat gain may be as great as 5000 Btu per hour or more per total name-plate horsepower. Of course, motors are not always fully loaded, nor do they always run continuously. Where considerable horsepower is involved, very careful survey data must be obtained in order to make an accurate cooling estimate. Relative to electric cooking equipment, the total heat figured is taken as 20 to 100 per cent of name-plate rating and the latent heat from 0 to 35 per cent of this total, depending on the use and type of appliance.

The total heat gain from gas-burning equipment is taken as 10 to 100 per cent of name-plate rating and the latent heat as 0 to 55 per cent of this total, depending on the use, type of appliance, and whether a gravity vent is used.

Steam-heated equipment also gives off heat, as do certain chemical processes and mechanical equipment, even though the source of power may be outside the space. These items occur mainly in industrial air-conditioning applications and require careful survey data.

When equipment is properly hooded and connected to a positive exhaust system, approximately half of the heat given off is carried away and approximately half of it gets into the room.

Values of heat liberated by appliances are given in Table 10.15.

10.11. Outside Air. In order to keep a room fresh, outside air is brought in to dilute and reduce the concentration of smoke, odors, carbon dioxide, or other undesirables. The number of people smoking and the ceiling height or volume of the space affect the volume of outside air required per unit time per person for air-conditioning applications. When local codes do not indicate specific quantities, values taken from Table 10.16 are recommended.

Certain fruits and vegetables are still alive when placed in a refrigerator and continue to mature. They require oxygen, and the carbon dioxide and the heat generated must be removed. In some cases outside air is introduced through the cooling apparatus. This has been done, particularly for apple, pear, and citrus-fruit storage, for banana ripening, and for ale and beer fermentation rooms. However, F. W. Allen, pomologist at the University of California, and W. T. Pentzer, plant physiologist, U.S. Department of Agriculture, state that outside-air ventilation is rarely practical for apples and pears.¹⁵

Both sensible heat and latent heat are added by the outside air, and in general these are calculated in the same manner as the infiltration components.

Many practicing engineers take into account coil "by-pass factors" when estimating the cooling load and supply-air quantity. The theory

¹⁵ From *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed., Chapter 13. New York: American Society of Refrigerating Engineers, 1946.

TABLE 10.15^{1a}
RATE OF HEAT GAIN FROM APPLIANCES WITHOUT HOODS^{a,b}

Appliance	Capacity	Over-all Dimensions (Less Legs and Handles; Last Dimension Is Height), Inches	Control A—Automatic M—Manual	Miscellaneous Data	Manufacturer's Rating	Maintaining Rate Btu per Hour	Recommended Rate of Heat Gain Btu per Hour		
							Sensible	Latent	Total
Restaurant Electrical Appliances									
Coffee Brewer and Warmer	$\frac{1}{2}$ gal								
Coffee Brewer Unit with Tank	$\frac{1}{2}$ gal	20 X 30 X 26	M M	Brewer 660 w Warmer 90 w	600° 90° 4960°	306	900 230 4800	220 60 1200	1120 290 6000
Coffee Urn	3 gal 5 gal	12 X 23 X 21 18 (Diam.) X 37	A A	Nickel plated Nickel plated	4500° 5000°	2600 3600	2200 3400	1500 2300	3700 5700
Doughnut Machine		22 X 22 X 57	A	Exhaust system	4700°		5000	0	5000
Egg Boiler	2 cups	10 X 13 X 25	M		1100°		1200	800	2000
Food Warmer, with Plate Warmer, per sq ft of top surface			A	Insulated, separate heat unit for each pot; plate warmer in base	400°	500	350	350	700
Food Warmer, alone, per sq ft of top surface			A		300°	400	200	350	550
Fry Kettle	11½ lb fat	12 (Diam.) X 14	A		2600°	1100	1600	2400	4000
Fry Kettle	25 lb fat	16 X 18 X 12	A	Area 12 X 14 in.	7000°	2000	3800	5700	9500
Griddle, Frying		18 X 18 X 8	A	Area 18 X 14 in.	2350°	2800	3100	1700	4800
Griddle, Frying		24 X 20 X 10	A	Area 23 X 18 in.	4000°	5000	5300	2900	8200
Grill, Meat		14 X 14 X 10	A	Area 10 X 12 in.	3000°	1900	3900	2100	6000

Grill, Sandwich		13 X 14 X 10	A	Area 12 X 12 in.	1650°	1900	2700	700	3400
Roll Warmer		23 X 23 X 29	A	Three drawers	1000°	900	2400	300	2700
Toaster, Continuous	360 slices/hr	15 X 15 X 28	A	2 slices wide	2200°	5000	5100	1300	6400
Toaster, Continuous	720 slices/hr	20 X 15 X 28	A	4 slices wide	3000°	6000	6100	2600	8700
Toaster, Pop-up	216 slices/hr	12 X 11 X 9	A	4 slice	2450°	2000	4900	900	5800
Waffle Iron	20 waffles/hr	12 X 13 X 10	A	7 in. diam. waffle	750°	600	1100	750	1850

Restaurant Gas-Burning Appliances

Coffee Brewer and Warmer	$\frac{1}{2}$ gal		M	Brewer Warmer	3400 ^d 500 ^d	500	1350 400	350 100	1700 500
Coffee Brewer Unit with Tank	4 $\frac{1}{2}$ gal Tank	19 X 30 X 26	M	4 Brewers and tank			7200	1800	9000
Coffee Urn	3 gal 5 gal	12 X 23 X 21 18 (Diam.) X 37	A A	Nickel plated Nickel plated		3400 4700	2500 3900	2500 3900	5000 7800
Food Warmer, per sq ft of top surface			M	Water bath	2000 ^d	900	850	430	1280
Fry Kettle	15 lb fat	12 X 20 X 18	A	Area 10 X 10	14250 ^d	3000	4200	2800	7000
Fry Kettle	28 lb fat	15 X 35 X 11	A	Area 11 X 16	24000 ^d	4500	7200	4800	12000
Grill		22 X 14 X 17	M	Insulated, grill surface of 1.4 sq ft Top burner 22,000 Btu/hr Bottom burner 15,000 Btu/hr	27000 ^d		14400	3600	18000

^a For restaurant appliances, miscellaneous electrical and miscellaneous gas burning appliances.

^b When these appliances are hooded and provided with adequate exhaust, use 50 per cent of recommended rate of heat gain from unhooded appliances.

^c Manufacturer's rating in watts.

^d Manufacturer's rating in Btu per hour.

¹⁶ From *Heating Ventilating Air Conditioning Guide 1948*, Chapter 15.

Hair Dryer, Helmet Type		M	Fan, 80 w; Low, 300 w; High, 710 w	705°	1870	330	2200
Permanent Wave Machine		M	60 heaters at 25 w each, 36 in normal use	1500°	850	150	1000
Neon Sign, per linear ft of tube			$\frac{1}{4}$ in. outside diam. $\frac{3}{8}$ in. outside diam.		30 60		30 60
Sterilizer, Instrument		A	For physicians; thermo- stat cuts off 550 w be- fore boiling	1100°	650	1200	1850

Miscellaneous Gas-Burning Appliances

Burners, Laboratory							
Small Bunsen	$\frac{7}{8}$ in. Carrel	M	Manufactured Gas	1800 ^d	960	240	1200
Small Bunsen	$\frac{1}{8}$ in. Barrel	M	Natural Gas	3000 ^d	1680	420	2100
Fishtail	$\frac{1}{8}$ in. Barrel	M	Manufactured Gas	3500 ^d	1960	490	2450
Fishtail	$\frac{1}{8}$ in. Barrel	M	Natural Gas	5500 ^d	3080	770	3850
Large Bunsen	1½ in. Mouth	M	Adjustable orifice	6000 ^d	3350	850	4200
Cigar Lighter		M	Continuous flame	2500 ^d	900	100	1000
Hair Dryer, 5 helmets		A	Heater and fan blowing air to helmets	33000 ^d	15000	4000	19000
Stoves, Oven			Insulated, modern	25000 ^d	7200	1800	9000
			Not insulated	25000 ^d	9200	2300	11500

^a For restaurant appliances, miscellaneous electrical and miscellaneous gas burning appliances.

^b When these appliances are hooded and provided with adequate exhaust, use 50 per cent of recommended rate of heat gain from unhooded appliances.

^c Manufacturer's rating in watts.

^d Manufacturer's rating in Btu per hour.

on which this practice is based is that the air in passing through a finned-tube coil does not all contact the coil or fin surface unless the coil is several rows deep in the direction of air flow. The by-pass factor is equal to the difference between the entering-air temperature and the mean surface temperature, divided by the difference between the leaving air temperature and the mean surface temperature.¹⁷ The by-pass factor for a coil of n rows deep equals the by-pass factor for one row raised to the n th power. If the by-pass factor for one row is 0.67 (ordinary air conditioning coil with about 7 fins per inch),

$$BF \text{ for 2 rows} = 0.67^2 = 0.45,$$

$$BF \text{ for 4 rows} = 0.67^4 = 0.20 \text{ and}$$

$$BF \text{ for 8 rows} = 0.67^8 = 0.04 \text{ (small enough to use zero).}$$

Therefore, for most coils 6 rows or less in depth, an appreciable part of the outside air slips past the coil uncooled and becomes part of the room

TABLE 10.16
VENTILATION REQUIREMENTS¹⁸

Smoking	No. of Occupants	CFM
None $\times 7\frac{1}{2} =$
Light $\times 15 =$
Heavy $\times 40 =$

load. The sensible and the latent heat gain of the outside air must each be divided, and the portions equivalent to the by-pass factor are added in with the room sensible and room latent heat loads, respectively (20 per cent for the 4-row coil above). The remainder of the outside-air heat gain is added to the total room load.

10.12. Miscellaneous Items. After the room sensible-heat components are totaled, a certain percentage of the sum is added to the total to account for supply-duct heat gain, supply-duct air leakage, supply-fan power input, and a safety factor when any of these items apply. This section presents one method of analysis by which these items may be computed.¹⁹

The supply-duct heat gain depends upon the temperature of the air in the duct and the temperature of the space surrounding the duct. If all of the duct is in the conditioned space, this item is zero; but for a long

¹⁷ Carrier, W. H., "The Contact-Mixture Analogy Applied to Heat Transfer with Mixtures of Air and Water Vapor." *Transactions A.S.M.E.*, Vol. 59, p. 49, 1937.

¹⁸ Reprinted, by permission, from *Application Engineering Standards for Air Conditioning for Comfort*, published by Air Conditioning and Refrigerating Machinery Association, Inc., 1947.

¹⁹ Carrier, Cherne and Grant, *Modern Heating, Ventilating and Air Conditioning*. New York: Pitman Publishing Corp., 1940, pp. 52-55.

run, less air than originally computed is required from the outlets near the fan, because of the panel cooling effect of the duct. More air than originally planned for is needed for the outlets near the end of the duct because of the rise in air temperature in the duct. If the duct is located in an unconditioned space, it usually must be insulated to prevent condensation. Insulation is also generally economical in order to reduce the duct heat gain.

The percentage to add to the room sensible heat to account for the supply-duct heat gain may be calculated from the following equation:

$$\text{Percentage gain} = 100 \times \text{length of duct} \times \text{temp. rise per unit length} \\ \times \frac{\text{air temp. outside duct} - \text{air temp. inside duct}}{\text{conditioned room temp.} - \text{fan discharge temp.}} \quad (10.16)$$

Values of temperature rise are given in Table 10.17. The figures for furred ducts are based on $\frac{3}{4}$ in. of metal lath and plaster and those for insulated ducts are based on 1 in. of cork, plastered.

TABLE 10.17
TRANSMISSION OF HEAT TO AIR DUCTS

CFM	Temperature Rise per F Difference per 100-ft Length		
	Uninsulated	Furred	Insulated
600	0.51	0.26	0.14
800	.46	.22	.12
1,000	.42	.20	.105
2,000	.30	.14	.070
4,000	.21	.094	.050
6,000	.17	.076	.042
8,000	.15	.060	.038
10,000	.14	.060	.035
20,000	.12	.050	.030

The loss due to supply-air leakage is not easy to estimate, since it depends upon the care used in the duct construction. Leakages as great as 30 per cent have been reported. The following are the recommended minimum percentages to add:

Long runs.....	10%
Medium runs.....	5%
Short runs.....	Neglect

The power consumed by the fan is converted into heat energy and imparted to the air. If the fan is on the entering side of the conditioning equipment, this energy must be added to the total load. If the fan is on

the leaving side, the heat is added to the room sensible-heat load. Values based on 50 per cent static efficiency and for various temperature differences between the conditioned room air and the supply air are given in Table 10.18.

The safety factor is sometimes called the factor of ignorance, and rightly so, since its value should be gaged by the accuracy or completeness of the survey data and possibly should include an allowance for unpredictable items. Values from 0 to 10 per cent are used in practice.

TABLE 10.18
HEAT DUE TO FAN HORSEPOWER

Total Fan Head in Inches Water	Per Cent of Room Sensible Heat					
	Fan Motor Within Conditioned Space or Air Stream			Fan Motor Not in Conditioned Space or Air Stream		
	Differential			Differential		
	10 F	20 F	30 F	10 F	20 F	30 F
0.75	6.5	3.5	2.5	5.5	3.0	2.0
1.25	11.0	5.5	3.5	9.5	4.5	3.0
1.75	15.5	7.5	5.5	13.0	6.5	4.5

The same percentages for supply-duct leakage loss and for safety factor as applied to the sensible-heat load are also applied to the sum of the room latent-heat loads, although the safety factor may be different in special cases.

The return-duct heat gain and air-leakage gain are generally not appreciable but should not be overlooked if the duct is long and must run through unconditioned space. Values from Table 10.17 may be used to compute the heat gain. These values, plus those for pump horsepower and dehumidifier and piping loss, if any, should be added to the total system load. Percentages for heat due to pump power are given in Table 10.19.

EXAMPLE 10.1. (a) Calculate the percentage to add to the room sensible heat for an insulated 60-ft supply-air duct in a 90 F space delivering 8000 cfm at 60 F to a room at 80 F.

$$\text{Percentage gain} = 100 \times 0.6 \times 0.038 \times \frac{90 - 60}{80 - 60} = 3.4\%$$

(b) Determine for the same system the heat due to fan power for a static pressure of 1.25 in. if the motor is not in the conditioned space.

$$80 - 60 = 20 \text{ F difference; from Table 10.18, gain} = 4.5\%$$

10.13. Cooling-Load Calculation. The principles and procedures explained in the preceding sections may be understood better by studying specific applications. Many manufacturers of air-conditioning or refrigerating equipment conduct training schools for their own and their dealers' engineers. Slight variations in certain details of calculating some of the items may be found when comparing the methods recommended by each concern, but in general the procedures are the same, and usually comparable equipment capacities are finally recommended for a given application.

TABLE 10.19
HEAT DUE TO PUMP HORSEPOWER²⁰

Pump Head, Ft.	Per Cent of Grand Total Heat Chilled Water Temperature Rise		
	5 F	7 F	10 F
35	2.0	1.5	1.0
70	3.5	2.5	2.0
100	5.5	4.0	2.5

It should be pointed out that the actual cooling load in a space is a function of time and of the heat capacity or storage effect of the contents and surrounding structure for the given space. Particularly careful consideration should be given to this fact in designing a system for a funeral home or other establishment having a peak load of comparatively short duration and considerably in excess of the usual load. Some of the phenomena of storage effect has been investigated in conjunction with recent studies made on panel cooling.²¹ Data on how to utilize this storage effect, however, are most difficult to find, and additional information would be a worth-while contribution to the profession. This storage effect may explain why certain systems with apparently undersized compressor capacity are operating fairly satisfactorily.

EXAMPLE 10.2. Calculate the cooling load for an optical-goods shop in Baltimore from the following data:

Total outside windows: 442 sq ft
 Net south glass: 80 sq ft with shade screen
 Net east glass: 180 sq ft with shade screen
 Somewhat hazy atmosphere outside

²⁰ Tables 10.17, 10.18, 10.19 are from Carrier, Cherne, and Grant, *Modern Heating, Ventilating and Air Conditioning*. New York: Pitman Publishing Corporation, 1940, pp. 53-54.

²¹ Leopold, C. S., "The Mechanism of Heat-Transfer-Panel Cooling Heat Storage." *Refrigerating Engineering*, Vol. 54, No. 1 (July, 1947), p. 33; Part II, Vol. 55, No. 6 (June, 1948), p. 571.

Walls: 16 in. brick

Net east: 288 sq ft

Net west: 0 sq ft

Net south: 137 sq ft

Net north: 370 sq ft

Floor area: 2000 sq ft

Ceiling height: 10 ft

Floor and ceiling: Double wood floor on wood joists, metal ceiling

Partition: 880 sq ft; Wood lath and plaster on both sides of studs.

Occupants: 25

Lights: 2500 watts

Motors: 15 hp, 64% efficient, fully loaded, running 50% of the time.

Inside design conditions: 80 F d.b., 67 F w.b.

By-pass factor for four-row coil: 0.2 in.

Figure 10.8 shows the total cooling load calculated from the above survey data. Determination of the apparatus dew point indicated is explained in Chapter 11.

10.14. Refrigerator Loads. In general, the heat sources in a refrigerator may be classified in accordance with the following items:

- (1) Heat transmission
- (2) Air leakage and ventilation
- (3) Product load
- (4) Miscellaneous internal sources

These items have been explained in the preceding sections, and a 10 per cent safety-factor item is added to their sum to allow for possible variations in the assumptions made. The last three items are often difficult to predict for refrigerators having a volume under 1600 cu ft. In such cases the load is divided into two components: transmission gain and usage load.

The transmission gain is calculated in the usual manner, but the door loss, product load, internal heat gain, and a safety allowance are based on a single factor determined from experience. These factors, indicating variation due to space volume, usage, and outside temperature minus room temperature, are shown in Table 10.20. Some estimators employ empirical rules for calculating loads based on refrigerator surface area only; others use refrigerator volume only. It is obvious that disagreeing answers would be obtained in many cases. The method described above is recommended.

Many estimators calculate an hourly load at the time of day when it will be a maximum. As the load drops off from this maximum, the compressor cycles or defrosting can be accomplished. Other estimators calculate a 24-hr load and then divide by a factor varying from 14 to 20 to arrive at a maximum hourly load for selecting equipment. The actual value used for operating hours per day is based on experience, allowing for a defrosting period and loading. Factors of 18 to 20 are used for installations with coil temperatures above freezing, 18 for room temper-

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<table border="1"> <thead> <tr> <th>Item</th> <th>Quantity</th> <th>Difference</th> <th>Factor</th> <th>Btu/Hour</th> </tr> </thead> <tbody> <tr> <td colspan="5">SUNLIGHT GAIN</td> </tr> <tr> <td>S Glass</td> <td>80 x .9 x .25</td> <td>Sq. Ft. x</td> <td>85</td> <td>1 600</td> </tr> <tr> <td>E Glass</td> <td>180 x .9 x .25</td> <td>Sq. Ft. x</td> <td>16</td> <td>600</td> </tr> <tr> <td>Skylight</td> <td></td> <td>Sq. Ft. x</td> <td></td> <td></td> </tr> <tr> <td>E Wall</td> <td>288</td> <td>Sq. Ft. x</td> <td>1.5</td> <td>400</td> </tr> <tr> <td>Other Wall</td> <td>507</td> <td>Sq. Ft. x</td> <td>1.5</td> <td>800</td> </tr> <tr> <td>Roof</td> <td></td> <td>Sq. Ft. x</td> <td></td> <td></td> </tr> <tr> <td colspan="5">TRANSMISSION GAIN</td> </tr> <tr> <td>All Glass</td> <td>442</td> <td>Sq. Ft. x</td> <td>1.13</td> <td>7 500</td> </tr> <tr> <td>Skylight</td> <td></td> <td>Sq. Ft. x</td> <td></td> <td></td> </tr> <tr> <td>Shade Wall</td> <td></td> <td>Sq. 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Ft. x	10	6 000											INFILTRATION AND OUTSIDE AIR					Infiltration		C.F.M. x	1.08		Outside Air	375	C.F.M. x	15 x .2 BF x 1.08	1 200	INTERNAL HEAT					People	25	People x	220	5 500	H. P.	.5 x 15	H. P. x	4000	30 000	Lights	2500	Watts x	3.4	8 500	Gas Burner				2 700						Room Sensible Heat Sub Total				73 800	Storage		Sq. Ft. x			Sub Total after Storage					Supply Duct	% + Supply Duct	% + Fan	5 % + Safety	0 %	Heat Gain	Leakage Loss	H. P.	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Ventilation = <u>375</u></p> <p>INFILTRATION</p> <p>Swinging Revolving Doors People x CFM/Per =</p> <p>Open Doors Doors x CFM/Door =</p> <p>Exhaust Fan Feet x CFM/Ft =</p> <p>Crack CFM. Infiltration =</p> <p>OUTSIDE AIR THROUGH APPARATUS</p> <p>DEHUMIDIFIED AIR</p> <p>(1 - .2 B.F.) x (80° Rm. T. - 68° App. D.P.) = <u>17.6</u> Dehum. Rise</p> <p>84900 Room Sens. = <u>4500</u> C.F.M.</p> <p>1.08 x 17.6 Dehumidifier Rise =</p> <p>Coordination of cooling design and heating design is essential and a study of the heating estimate should be made at this time.</p> <p>NOTES</p> <p>Outside metal shading screen; 0.25.</p> <p>Somewhat hazy atmosphere; 0.95.</p> <p>Loads calculated for 10 A.M. and for 12 A.M. are less than shown.</p> <p>15 hp in motors averaging 64% efficiency fully loaded and on 50% of the time.</p>				
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Fig. 10.8. Cooling-estimate form. From Heating Piping and Air Conditioning Contractors National Association, *Engineering Standards, Part IV*, 1944.

atures below zero, and 14 to 16 for intermediate temperatures when frequent defrosting is recommended.

EXAMPLE 10.3. (a) Calculate the cooling load for a room 14 × 26 × 11 ft high at 35 F designed only to chill 12,000 lb of beef from 100 to 45 F in 24 hr. There are to be 2 occupants and 600 watts of light. Outside temperature is 95 F.

Wall:	$154 \times 10 \times 0.15$	=	230	(with 2 in. cork insulation)
Wall:	286×0			
Wall:	$154 \times 55 \times 0.08$	=	680	(with 4 in. cork insulation)
Wall:	$286 \times 70 \times 0.06$	=	1,200	(with 5 in. cork insulation)
Floor:	$364 \times 50 \times 0.08$	=	1,460	(with 4 in. cork insulation)
Ceiling:	$364 \times 90 \times 0.05$	=	1,640	(with 6 in. cork insulation)
Door loss:	$4000 \times 60 \times 0.008$	=	1,920	
People:	2×760	=	1,520	
Lights:	600×3.4	=	2,040	
Product:	$\frac{12,000 \times 55 \times 0.75}{24 \times 0.8}$	=	25,900	
			<u>36,590</u>	
	Plus 10%	=	<u>3,660</u>	
			40,250	Btu per hr

(b) Calculate the cooling load for a refrigerator $9 \times 4 \times 8$ ft high, used for vegetable service, at 40 F and 90 per cent relative humidity. Outside room temperatures are 95 F dry bulb and 78 F wet bulb.

Walls:	$208 \times 55 \times 0.10$	=	1150	3 in. cork
Floor:	$36 \times 55 \times 0.10$	=	200	
Ceiling:	$36 \times 55 \times 0.10$	=	200	
Usage:	$288 \times \frac{35}{24} \times 0.075 \times (41.4 - 14.7)$	=	840	
			<u>2390</u>	
	Plus 10%	=	<u>240</u>	
			2630	Btu per hr

$$h_o = 0.24 \times 95 + \frac{118}{7000}(1061 \times 0.444 \times 95) = 41.4 \text{ Btu per lb}$$

$$h_i = 0.24 \times 40 + \frac{33}{7000}(1061 \times 0.444 \times 40) = 14.7 \text{ Btu per lb}$$

10.15. Fluid Cooling Loads. When refrigeration applications involve the direct cooling of a fluid, the amount of refrigeration required, or total load, usually consists of two components: fluid cooling and equipment losses.

The amount or rate of flow of fluid; its specific heat in the cooling range, its initial temperature, and its desired final temperature must be determined. These are combined to give the cooling load, of the fluid only, in the equation

$$q = \text{lb per hr} \times \text{sp ht} \times (t_1 - t_2) \text{ Btu per hour} \quad (10.17)$$

Of course the heat exchanger must be properly designed so that the fluid can be cooled as desired. Even so, there often are heat gains from surroundings, for which an allowance should be made. In some instances this allowance can be calculated. In other cases it should be based on experience and good engineering judgment.

Whenever a change of state of the fluid takes place, the latent heat of condensation or of freezing must be considered.

TABLE 10.20
USAGE HEAT LOSS, BTU PER 24 HR FOR ONE CU FT INTERIOR CAPACITY²²

Volume cu ft	Service	Temperature reduction in F (Room temp. minus box temp.)								
		40	45	50	55	60	65	70	75	80
15	Normal	108	122	135	149	162	176	189	203	216
	Heavy	134	151	168	184	201	218	235	251	268
50	Normal	97	109	121	133	145	157	169	182	194
	Heavy	124	140	155	171	186	202	217	233	248
100	Normal	85	96	107	117	128	138	149	160	170
	Heavy	114	128	143	157	171	185	200	214	228
200	Normal	74	83	93	102	111	120	130	139	148
	Heavy	104	117	130	143	156	169	183	195	208
300	Normal	68	77	85	94	102	111	119	128	136
	Heavy	98	110	123	135	147	159	172	184	196
400	Normal	65	73	81	89	97	105	113	122	130
	Heavy	95	107	119	130	142	154	166	178	190
600	Normal	61	68	76	84	91	99	106	114	122
	Heavy	91	103	114	125	137	148	160	171	182
800	Normal	59	67	74	81	89	96	104	111	118
	Heavy	89	100	112	123	134	145	156	167	178
1,000	Normal	57	64	72	79	86	93	100	107	114
	Heavy	86	97	108	119	130	140	151	162	173
1,200	Normal	55	62	69	76	83	90	97	104	110
	Heavy	84	95	105	116	126	137	147	158	168
1,600	Normal	51	58	64	70	77	83	90	96	102
	Heavy	79	89	99	108	118	127	138	148	158

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²² From *Refrigerating Data Book*, 5th ed., 1943. New York: American Society of Refrigerating Engineers, p. 173.

4. *Heating and Ventilating*, "H. & V.'s Reference Data—241 & 242," Vol. 40, No. 3 (March, 1943).
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PROBLEMS

10.1. Calculate the cooling load for the following office in Chicago. Inside conditions 78 F dry bulb, 50 per cent relative humidity. Exposed walls on the south and west are 60 ft and 40 ft long, respectively. Ceiling height is 10 ft. Windows are single glass with venetian blinds, 180 sq ft in south wall, 120 sq ft in west wall. Walls are of 8 in. brick, furred and plastered, the east and north partitions are of 4 in. hollow tile with metal lath and plaster on each side. Ceiling U is 0.15. The space below the office is at 78 F. There are 4800 watts of light and 30 occupants. Spaces above and adjoining partitions are at 90 F.

10.2. Calculate the cooling load for a freezing room $6 \times 8 \times 8$ ft high at 0 F if 400 lb of poultry is to be chilled from 40 to 10 F in 20 hr. There are 200 watts of light. Recommended minimum insulation is used and the outside temperature is 80 F.

10.3. Calculate the cooling load for a refrigerator $8 \times 5 \times 8$ ft high used for general heavy service. Inside conditions are to be 36 F and 85 per cent relative humidity. Outside conditions are 90 F dry bulb and 75 F wet bulb. Recommended minimum insulation is used.

10.4. (a) Calculate the load for an apple storage room $10 \times 30 \times 8$ ft high at 40 F and holding 8000 lb of apples. Outside temperature is 80 F and normal insulation is used. (b) Calculate the load for the same room if 1000 lb is to be cooled from 80 F to 40 F in 48 hr.

10.5. Calculate the Btu per hour heat gain to an insulated duct that carries 4000 cfm for 30 ft and then divides into two ducts carrying 2000 cfm each, one for 10 ft and the other for 30 ft. The air at the start of the duct is 60 F. The surrounding space is at 100 F.

10.6. A restaurant at 40° latitude has a total heat gain of 9.5 tons at 12 noon. The southern exposure consists of 200 sq ft of unshaded glass. What would be the total load if (a) inside Venetian blinds are added and (b) if an awning is added? The miscellaneous room sensible heat gain items total 10 per cent.

10.7. How many tons of refrigeration are required to cool 50 gpm of oil having a specific gravity of 0.83 and a specific heat of 0.51 from 80 F to 60 F?

10.8. Calculate the room sensible and latent heat loads for a celery storage room $19 \times 25 \times 10$ ft high at 32 F and 90 per cent humidity using 4-in. insulation, when the outside conditions are 90 F and 60 per cent humidity. Twenty tons of celery is received periodically and should be chilled 5 F in 24 hr. The product latent heat, 0.05 Btu per pound per hour, is included in the total product cooling load. There are 300 watts of light and 1 person. Add 10 per cent to sensible heat gain for fan heat and unknown items.

10.9. Calculate the air-conditioning load at 80 F and 50 per cent humidity for a jewelry store in Baltimore, Md., having 72 sq ft of south glass with inside Venetian blinds, 156 sq ft total glass, 45 sq ft of show-window partition, 123 sq ft of 12 in. brick south wall, 1440 sq ft of 4-in. tile-and-plaster partition, and 700 sq ft of plaster ceiling under an uninsulated 1-in. wood-plus-asphalt roof. There are 14 occupants and 2800 watts of light.

10.10. Calculate the load for a freezer room $11 \times 11 \times 11$ ft high at 0 F if the temperature is 95 F, 90 F, 40 F, and 0 F respectively outside opposite each side. The room is on the ground and there is a room at 90 F above. Thirteen hundred pounds of meat, half beef and half pork, is to be cooled from 40 to 0 F in 20 hr. Recommended minimum insulated construction is used throughout. The ground temperature is 50 F.

CHAPTER 11

Condensers and Evaporators

11.1. Functions. In designing a refrigeration system, the next logical step after determining the cooling load is to select the proper equipment. The components to consider are the compressor, condenser, evaporator, piping, and controls. Compressors were discussed in Chapter 7. This chapter will deal with condensers and evaporators, both being heat exchangers with certain features in common. Piping and controls will be covered in Chapters 12 and 14.

As the name implies, a condenser should be designed to condense effectively the compressed refrigerant vapor. It is in the condenser that the refrigerant must give up the heat absorbed in the evaporator plus the heat added by the compressor. Good design further provides for some subcooling of the liquid refrigerant before it leaves the condenser. The pressure of the refrigerant in the condenser is very little less than at the compressor discharge, where it is higher than at any other part of the system. Hence the condenser and other equipment in the system between the compressor discharge and the expansion valve are often referred to as *high-side equipment*.

The evaporator is that part of the system in which the refrigerant evaporates or boils; it is the cooling unit, where heat is absorbed by the refrigerant. A small part of the liquid refrigerant flashes in passing through the expansion valve or other restriction used where the greatest pressure drop from condenser to evaporator takes place. However, most of the refrigerant entering the evaporator is in the liquid state, and *dry expansion* design provides for complete evaporation and for the vapor to leave the evaporator slightly superheated. In *flooded evaporators* all the refrigerant is not evaporated; the liquid-vapor mixture leaving the evaporator flows into a surge drum from which the vapor is drawn off into the compressor suction line and the liquid is recirculated through the evaporator after mixing with additional liquid from the receiver.

Since the refrigerant in the evaporator is at a low pressure compared to that in the condenser, the evaporator and the auxiliaries between the expansion valve and the compressor intake are often referred to as *low-side equipment*.

11.2. Condenser Theory. As discussed in §8.16, there are two types of condensation, film and dropwise.

Dropwise condensation occurs only under special conditions, as on a polished surface very lightly coated with a nonwetting agent. Hydro-

carbon oils will serve such a purpose only temporarily. More successful agents include oleic acid when applied to copper, brass, nickel, and chromium, and mercaptans (alcohol-type compounds with sulfur replacing the oxygen) on copper or its alloys.¹ Most evidence of dropwise condensation has been noted for nonmiscible vapor mixtures; steam is the only pure vapor reported on extensively.

Dropwise condensation is desirable, since heat-transfer coefficients of four to eight times those for the film type have been obtained. These higher values logically should be expected because of the nature of the process. Droplets form and grow until they are heavy enough to run off and in so doing do not wet the surface. A large part of the surface is dry at any one time, thus allowing direct contact with the vapor and therefore better heat transfer than is possible with an insulating liquid film on the surface.

Film-type condensation is much more common than dropwise, and equation 8.21 derived by Nusselt then applies.² This equation is conservative if the vapor is turbulent, and test values higher than theoretical have been reported. If the constants given for equation 8.21 are used and the expression for a vertical tube is equated to that for a horizontal tube with the variables kept constant, the heat-transfer coefficient is the same for each tube position when the length of the tube is 2.87 times the outside diameter. Since it is practical to have condenser-tubes longer than this, most condensers are horizontal to take advantage of the better heat transfer. With a tube length 100 times the diameter, the coefficient for a horizontal tube would be 2.43 times that for a vertical tube. In the case of a bank of horizontal tubes, D should be taken as the sum of the diameters of the tubes in a vertical row.

In 1915 Wilson used an effective graphical analysis of over-all heat-transfer coefficient tests.³ This method is illustrated by McAdams⁴ It utilizes the concept that the reciprocal of U is equal to the sum of the individual heat resistances of the vapor-side film, the wall, the dirt or scale deposit or other fouling effect, and the water-side film. A rectangular coordinate plot of the over-all resistance $1/U$ against the reciprocal of the water velocity to the 0.8 power produces a straight line for comparable test data on a condenser. An equation for this line can be written in the form

$$\frac{1}{U} = A + \frac{1}{B(V)^{0.8}} \quad (11.1)$$

¹ McAdams, W. H., *Heat Transmission*, 2d ed. New York: McGraw-Hill Book Company, Inc., 1942, p. 276.

² A derivation is presented in McAdams' *Heat Transmission*, p. 260.

³ Wilson, E. E., "A Basis for Rational Design of Heat-Transfer Apparatus." *Transactions ASME*, Vol. 37 (1915), p. 47.

⁴ McAdams, W. H., *op. cit.*, pp. 271-275.

The constant A is equal to the sum of the resistances of the vapor-side film, the wall, and the fouling effect; and B is an empirical constant. The resistance of the wall can be calculated, so that only the vapor-side film and fouling-effect values remain unknown. Making tests on clean tubes would leave only the vapor-side film resistance, and the difference in A values for clean and dirty tubes would be the apparent resistance of the fouling. Equation 11.1 applies for turbulent water flow; for streamline flow $1/V$ should be raised to the $\frac{1}{2}$ power instead of to 0.8.

11.3. Condenser Design. There are three general types of refrigeration condensers: air-cooled, water-cooled, and a combination air- and water-cooled usually referred to as an evaporative condenser. Details of each of these are described in the following sections.

The type and size of condenser selected for a given system are based upon the economics involved. A desirable balance of initial and operating costs must be determined from the costs of power, water, condenser surface, and other equipment required. The size of a particular type of condenser chosen for a given load depends upon (1) the heat-transfer characteristics of the surface, of the refrigerant, and of the cooling medium; (2) the temperatures of the refrigerant and of the cooling mediums, and (3) the physical properties of the refrigerant. Allowances should be made for dirt and scale collection on the surfaces when laboratory heat-transfer test values are used in rating condensers. Capacity tables of reliable manufacturers allow for such items.

When air-cooled or water-cooled condensers of under 50 tons capacity are used, they usually are mounted on the same base as the compressor. The entire assembly is then referred to as an air-cooled or a water-cooled condensing unit. •

The amount of heat removed in the condenser is equal to the rate of refrigerant flow times the difference between the enthalpy values of the entering and leaving refrigerant. The amount of air or water required for condensing is equal to the total heat to be removed divided by the difference between the enthalpy values of the leaving and entering cooling fluid used.

Although low condensing pressures are economically desirable, the leaving-refrigerant pressure must be great enough to overcome the static head and the resistance of the mains, fittings, and controls. Too low a condenser pressure can cause trouble, and therefore automatic control of the condenser cooling medium is desired to give satisfactory operation during load variations.

11.4. Air-cooled Condensers. Air-cooled condensers are used in systems of the smallest capacity, up to 2 or 3 tons. A few larger units, up to 10 tons capacity, have been made but have proved impractical because of the large amount of surface required. However, condensing units of 1 hp and under are predominately air-cooled unless the ambient-

air temperature is unusually high. Their main advantages are simplicity and low installation cost. Since the only external connection required is to a power source, they are used in most self-contained units such as domestic refrigerators, freezer cabinets, display cases, water coolers, and room air conditioners.

A typical air-cooled condensing unit is shown in Fig. 11.1. The condenser consists of a finned continuous-tube coil with the compressor

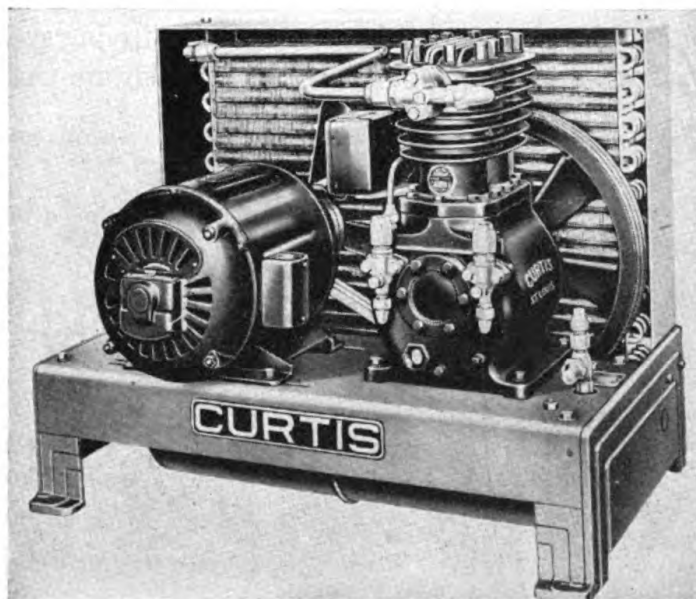


Fig. 11.1. Air-cooled condensing unit. Courtesy Curtis Manufacturing Co.

discharge vapor entering at the top and liquid leaving at the bottom, from where it flows into the receiver located under the base. A fan mounted on the compressor-motor pulley blows air across the condenser coil. The blade-shaped flywheel spokes aid air circulation across the condenser and compressor. The unit should be located in a well-ventilated and preferably cool space. The lower the ambient air temperature, the less power required for a given capacity.

A hermetically sealed compressor with an air-cooled condenser is shown in Fig. 11.2. This unit requires a separate motor for the condenser fan. Note that the receiver is vertical and is mounted above the base in order to save space.

11.5. Water-cooled Condensers. Water-cooled condensers are used with compressors of 1 hp and larger. They usually constitute the most economical choice of condenser where an adequate supply of clean inexpensive water of minimum corrosiveness is available together with adequate and inexpensive means of water disposal. They are used almost exclusively on large installations involving units of 100 hp. However, in these cases the amount of water needed generally warrants the installa-

tion of a cooling tower. An exception would be an installation adjacent to a clean river or a lake from which the water could be pumped and then returned. The condensers themselves may be of several types.

Double-pipe condensers, seldom installed nowadays, consist of a pipe within a pipe or a tube within a tube. Units fabricated from 1½-in. and 2-in. steel pipe in 10- to 20-ft lengths have been used in many ammonia systems. The tube sections are placed in a horizontal position to form a vertical bank, and either fittings or welded construction may be used. The complete unit can then be mounted on the wall, occupying space that often is of comparatively little value. Water usually enters the bottom sec-

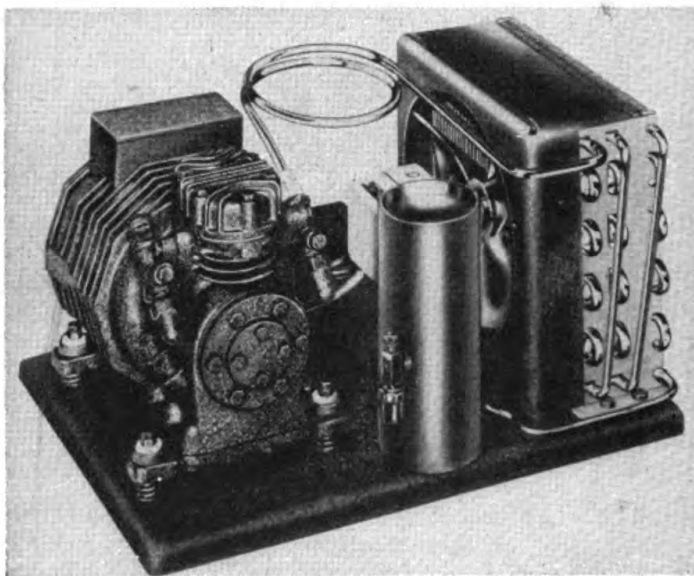


Fig. 11.2. Air-cooled hermetically sealed condensing unit. Courtesy Copeland Refrigeration Corp.

tions, flowing upward through the inner pipe. The compressed vapor enters the top section, flowing downward, thus providing counterflow. Several tests made on this type of condenser indicate that the heat-transfer coefficient is a function of the water velocity and of the mean temperature difference.^{5,6} Values recommended for the prediction of actual performance are shown in Fig. 11.3.

Double copper-tube condensers are used on some Freon-12 and methyl chloride condensing units. A typical unit of this type is shown in Fig. 11.4. The receiver is located under the base.

Shell-and-coil condensers are used on many modern water-cooled condensing units of 50 hp and less. The condenser is usually suspended

⁵ Kratz, A. P., et al., *University of Illinois Engineering Experiment Station Bulletins* 171 (1927), 186 (1928), and 209 (1930).

⁶ Stewart, F. E., and Holland, A. D., "Double Pipe Cooler and Condenser Tests." *Refrigerating Engineering*, Vol. 17, No. 1 (January, 1929), pp. 5-15.

under the base of the unit and also serves as the receiver. It consists of a welded steel shell with a continuous copper-tube coil through which the water circulates. The compressed refrigerant vapor is discharged into the shell, comes in contact with the cool coil, and condenses, and the liquid collects at the bottom.

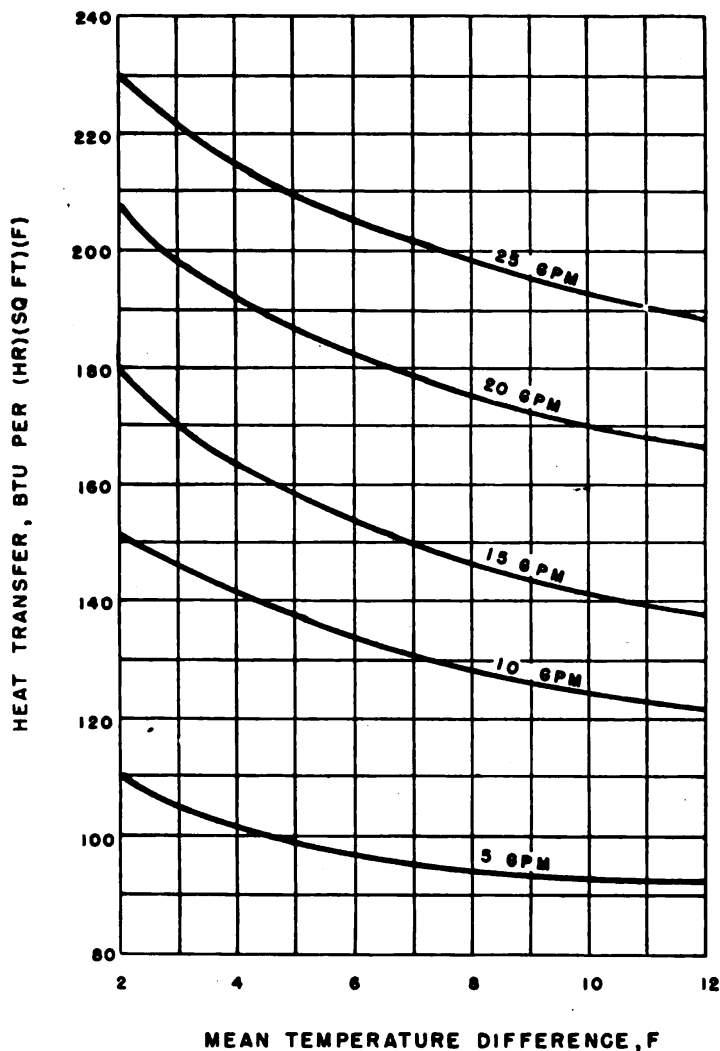


Fig. 11.3. Heat-transfer values for double-pipe condensers ($1\frac{1}{4}'' \times 2''$). From *Refrigerating Data Book*, 5th ed. 1943. American Society of Refrigerating Engineers, p. 263.

A shell-and-coil condenser is comparatively simple to construct. Its main disadvantage is that the water coil is difficult to clean. Should a leak develop in the coil, the shell must be opened and the entire coil removed in order to be repaired. A condensing unit with a shell-and-coil condenser is shown in Fig. 11.5.

Shell-and-tube condensers are used on most large modern installations and with the smaller compressors when fouling of the water circuit is

possible. The manufacturing cost, greater than for a shell-and-coil type, may be offset by reduced servicing costs.

A shell-and-tube condenser is similar to that generally employed for condensing steam. It consists of a shell with a tube sheet securely fas-

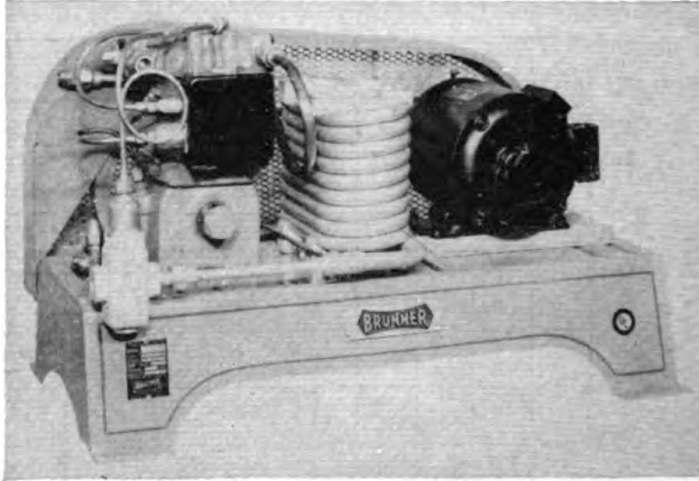


Fig. 11.4. Double-tube water-condensing unit. Courtesy Brunner Manufacturing Co.

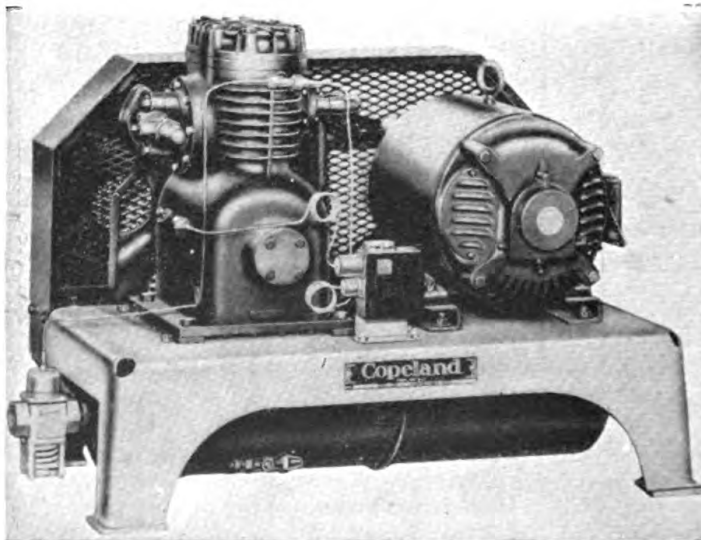


Fig. 11.5. Shell-and-coil water-condensing unit. Courtesy Copeland Refrigeration Corp.

tened to each end. Plain tubes or tubes with small fins formed on them are inserted through the holes in the tube sheets. The ends of each tube are rolled into the tube sheets to make a smooth, tight connection. Each tube sheet has an easily removable header-type cover, which is often baffled to cause the water to flow back and forth through the shell several times. This baffling increases the water velocity for a given flow and in turn increases the heat-transfer coefficient. A condensing unit with a

shell-and-tube condenser is shown in Fig. 11.6, and a typical large condenser of this type is shown in Fig. 11.7.

Coefficients for condensing Freon-12 on a plain $\frac{3}{4}$ -in. tube and on integral finned $\frac{3}{8}$ -, $\frac{3}{4}$ -, and 1-in. outside or root diameter single tubes were reported⁷ in 1947. Tubes with fin heights roughly 0.05 and 0.33 in.

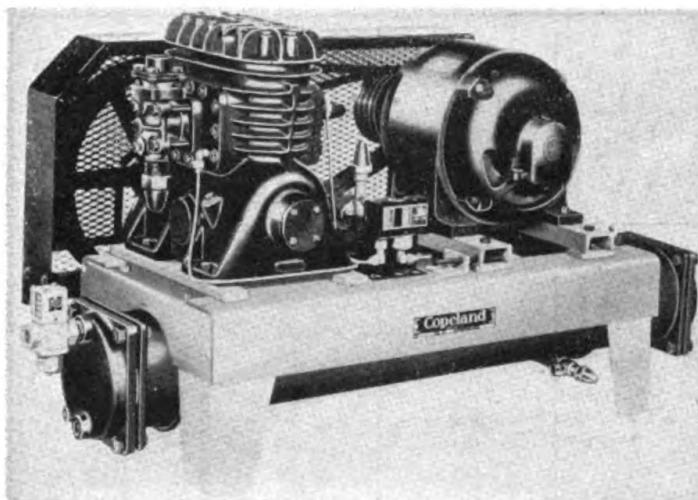


Fig. 11.6. Shell-and-tube water-condensing unit. Courtesy Copeland Refrigeration Corp.

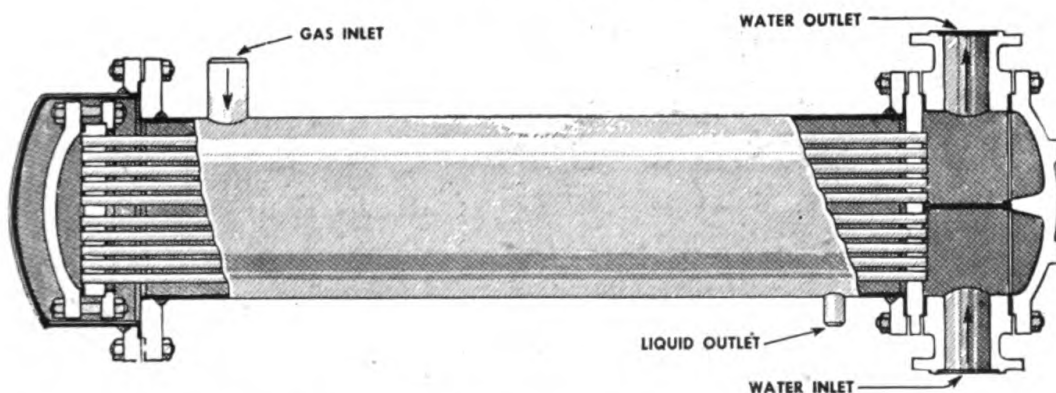


Fig. 11.7. Two-pass shell-and-tube condenser. Courtesy Worthington Pump and Machinery Corp.

were used. The values obtained are compared in Table 11.1 with those computed from equation 8.21, applying the constant for horizontal surfaces to the exposed tube area and the constant for vertical surfaces to the exposed fin area. Experimental values obtained for the water film coefficient on the inside of the tubes were 15 to 35 per cent higher than those computed from equation 8.14. This increase was attributed in part to the turbulence of the entering water and also the internal tube

⁷ Katz, Donald L., et al., "Condensation of Freon-12 with Finned Tubes." *Refrigerating Engineering*, Vol. 53, No. 3 (March, 1947), p. 211.

grooves made when forming the integral fins. These values must be modified for multitube condensers because of the flow of liquid condensed on the upper tubes. A second publication by the same authors covers

TABLE 11.1
COMPARISON OF EXPERIMENTAL VALUES FOR FILM COEFFICIENTS WITH CALCULATED VALUES⁸

Root Diameter, in.	No. of Fins per in.	Freon Temp., F	Average Temperature Difference, F	Exper. Film Coefficient (incl. Copper) Btu/hr (F) (sq ft)	Calculated Film Coefficient Btu/hr (F) (sq ft)
3/4	..	90	21	286	306
3/4	..	115	65	208	237
3/4	16	90	40	425	430
3/4	16	90	36	413	419
3/4	16	105	52	370	377
1	16	90	40	377	399
3/4	7	90	40	215	291
3/4	6	90	40	312	290
3/4	4	90	40	222	287

multitube condensers.⁹ A sample integral finned tube is shown in Fig. 11.8. However, shorter fin heights than shown are usually used on condenser tubes.

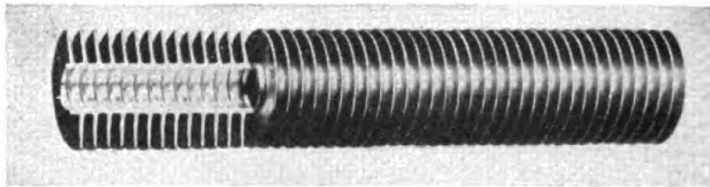


Fig. 11.8. Integral-fin tubing. Courtesy Wolverine Tube Division.

11.6. Economical Water Rate. The lower the water quantity used, the higher is the condensing temperature and the greater are the power costs. Conversely, the higher the water quantity used, the lower will be the condensing temperature and the smaller the power costs. There is, therefore, an optimum water flow that will produce a minimum sum of water and power costs. If power costs \$*A* per horsepower-hour, if water costs \$*B* per 1000 gal, and if the increase in horsepower per degree F rise in condensing temperature per ton of refrigeration is *m*, then the total cost

⁸ From Katz, Donald L., *et al.*, "Condensation of Freon-12 with Finned Tubes." *Refrigerating Engineering*, Vol. 53, No. 3 (March, 1947), p. 211.

⁹ Katz, Donald L., *et al.*, "Condensation of Freon-12 with Finned Tubes." *Refrigerating Engineering*, Vol. 53, No. 4 (April, 1947), p. 315.

of power for a temperature increment t_d is Amt_d/η , where η is the combined efficiencies of the compressor and motor. The cost of the condenser water per ton of refrigeration per hour is $60HB/(100t_d8.33) = 0.0072BH/t_d$, where H is the heat removed from the refrigerant per ton per minute.¹⁰ The total of the power and water costs are

$$C = \frac{Amt_d}{\eta} + \frac{0.0072BH}{t_d} \quad (11.2)$$

The minimum cost is then found by equating the first derivative with respect to the temperature increment equal to zero,

$$\frac{dC}{dt} = \frac{Am}{\eta} - \frac{0.0072BH}{t_d^2} = 0$$

or

$$t_d = 0.0848 \left(\frac{B\eta}{A} \right)^{\frac{1}{2}} \left(\frac{H}{m} \right)^{\frac{1}{2}} \quad (11.3)$$

m is determined by using equation 4.14.

The economical water rate is then

$$\text{gpm} = \frac{H}{8.33t_d} \quad (11.4)$$

Another method of determining minimum operating costs has been proposed by Boehmer.¹¹ Figure 11.9 shows a chart presented for determining the most economical condensing temperature. The minimum cost of operation may be determined for units of 10 to 60 hp from the equation

$$C = E(1.19r^{0.48} - 1.0) + \frac{M(146 + 55.6r^{0.48})}{t_c - 5 - t_1} \quad (11.5)$$

where C = total operating cost, cents per hr (ton)

E = electric rate, cents per kw hr

r = ratio of condensing pressure to suction pressure

M = water rate, cents per cu ft

t_c = condensing temperature, F

t_1 = entering water temperature, F.

11.7. Evaporative Condensers. This type of condenser was developed extensively in the early 1930's to alleviate the problems arising from the use of numerous water-cooled condensers in small air-conditioning systems. In many communities the water supply and drainage facilities

¹⁰ Macintire, H. J., *Refrigerating Engineering*. New York: John Wiley & Sons, Inc., 1940, pp. 181-182.

¹¹ Boehmer, A. P., "Condensing Pressures for Air Conditioning." *Heating, Piping and Air Conditioning*, Vol. 18, Nos. 9, 10 (September, October, 1946), pp. 77, 94.

were becoming overburdened. In some localities the high cost of water was a serious objection to the extensive use of air conditioning, and the use of a cooling tower for small installations was not practical. Therefore the evaporative condenser was designed to combine the functions of a condenser and a cooling tower.

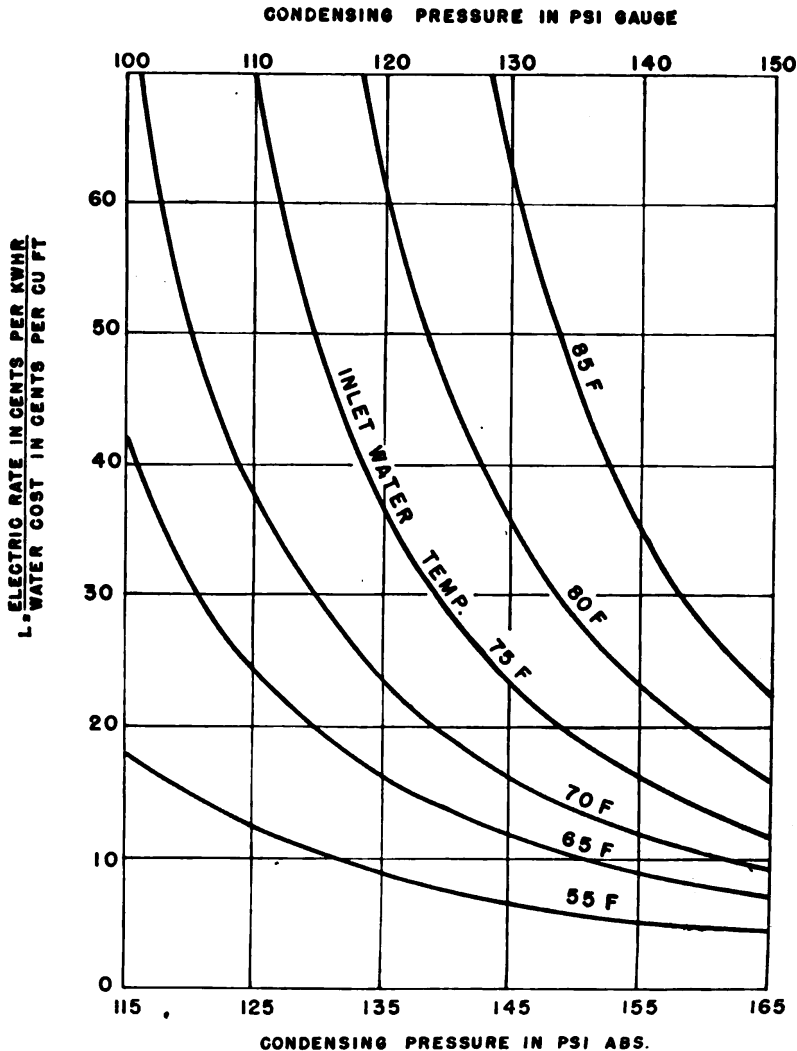


Fig. 11.9. Economical condensing-temperature chart. From *Heating, Piping and Air Conditioning* (Oct., 1946), by A. P. Boehmer.

Figure 11.10 is a diagrammatic sketch showing how a typical unit functions. Air is drawn in through an opening near the bottom, flows upward across the refrigerant coil, through the sprays and eliminators, and into the fan, and is discharged at the top of the unit. The refrigerant condensing coil is usually of the extended surface or finned type. The refrigerant enters at the top of the coils and flows across and downward. The condensed liquid drains into the receiver, frequently located in the

water tank in order to subcool the liquid refrigerant further. Water from the tank is withdrawn by a pump mounted on the unit and is discharged through spray nozzles directed downward over the refrigerant coil. Since 3 to 5 per cent of the water circulated evaporates, make-up water is admitted to the tank through a float-operated valve. The quantity needed is from 5 to 10 per cent of the water required for comparable water-cooled condensers. Partial evaporation increases the salt concentration of the water, and therefore treatment of the water and a continuous overflow (about $\frac{1}{8}$ gpm per ton) is recommended. Water treatment should also be used to reduce scale formation on the warm coil. Centrifugal fans are used except for the very small units. Large multiple units have been developed and used for capacities of well over 100 tons.

Units may be located in equipment rooms or other indoor spaces where they may serve as exhaust units for the rooms. They are also made weatherproof for outside locations. When they are so placed, provision must be made for draining if subfreezing temperatures occur. In cool weather it may not be necessary to use any water, and the unit then operates as an air-cooled condenser.

Capacities of evaporative condensers are difficult to calculate, but ratings of reliable manufacturers may be used. Since the entering-air wet-bulb temperature affects the refrigerant condensing temperature, the unit and the compressor used should both be selected to handle the design load at a wet-bulb temperature 2 to 3 deg above the design wet-bulb temperature.

11.8. Cooling Towers and Spray Ponds. Cooling towers and spray ponds are used when water-cooled condensers are desired and water is scarce or expensive. Here again an economic study should be made to determine whether the savings in water-supply cost over a reasonable length of time would warrant the additional initial and maintenance costs of a cooling tower or similar device. Evaporative condensers are more economical than cooling towers for most reciprocating-compressor installations up to 100 tons if the condenser can be located close to the compressor. Cooling towers are used, however, for larger installations, absorption refrigeration units, steam-jet systems, and centrifugal refrigeration units.

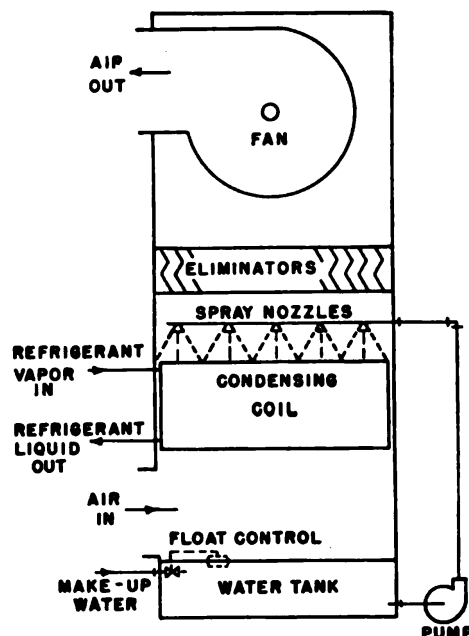


Fig. 11.10. Evaporative condenser.

When air is used to cool water, the rate of heat transfer depends upon (1) the difference between the air wet-bulb temperature and the water temperature, (2) the area of water surface exposed to the air, and (3) the relative velocity of the air and water. Secondary factors also influence the performance of various types of systems. The size of equipment required for the cooling of a given water quantity is affected by (1) the cooling temperature range of the water, (2) the difference between entering-air wet-bulb and leaving-water temperature, (3) design wet-bulb temperature itself, and (4) time of contact of air and water.

Natural ponds were first used with water discharged near the shore and the intake water taken from the center and as far below the surface as possible. In order to provide a greater area of exposed water, spray ponds were developed. For these, nozzles are located so as to spray the discharge water into the air. They should break up the water into droplets but not into a mist, which would easily drift off. In addition to providing more water-surface area, the water droplets travel with greater velocity than the surface water on a pool, the air velocity is greater at a distance above the pool than at the pool surface, and a greater volume of air contacts the water.

Nozzles are generally located from 5 to 7 ft above the surface, delivering 20 to 60 gpm at 5 to 12 psi and spaced to deliver 0.5 to 0.7 gpm per square foot of pool surface. Nozzles should not be located less than 20 ft from the edge of the pool, and even then louvered fences should be constructed at the edge to prevent entrained water from drifting beyond the pool. Potassium permanganate may be added to prevent growth of algae in warm weather. Ponds are located on the ground or on treated roofs. In many instances sufficient horizontal area is not available, and cooling towers are used.

Cooling towers may be classified as atmospheric or natural draft of the spray type and the deck type, and as mechanical draft of the forced type and the induced type.

Spray-type natural-draft towers are generally more economical for loads under 30,000 Btu per minute. They are from 6 to 15 ft high, actually constituting narrow spray ponds with a high louvered fence. They occupy less space than a pond and deliver from 0.6 to 1.5 gpm per square foot of base area. The sprayed louvers present additional exposed water surface. A typical atmospheric spray tower is shown in Fig. 11.11.

Deck-type natural-draft towers are also called atmospheric towers. They are from 20 to 40 ft high and 10 to 20 ft wide. The length depends upon the capacity required. They are made of wood or metal, with open and staggered horizontal decks at regular intervals from top to bottom. The water is discharged at the top and overflows from deck to deck to a basin at the bottom. Wind splashboards usually extend outward and upward around the outside. These towers must be designed for wind

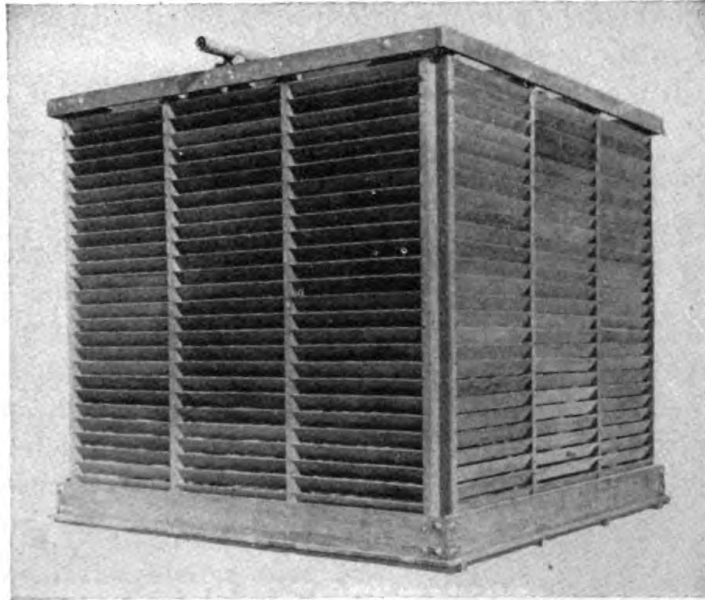


Fig. 11.11. Atmospheric spray tower. Courtesy The Marley Co., Inc.

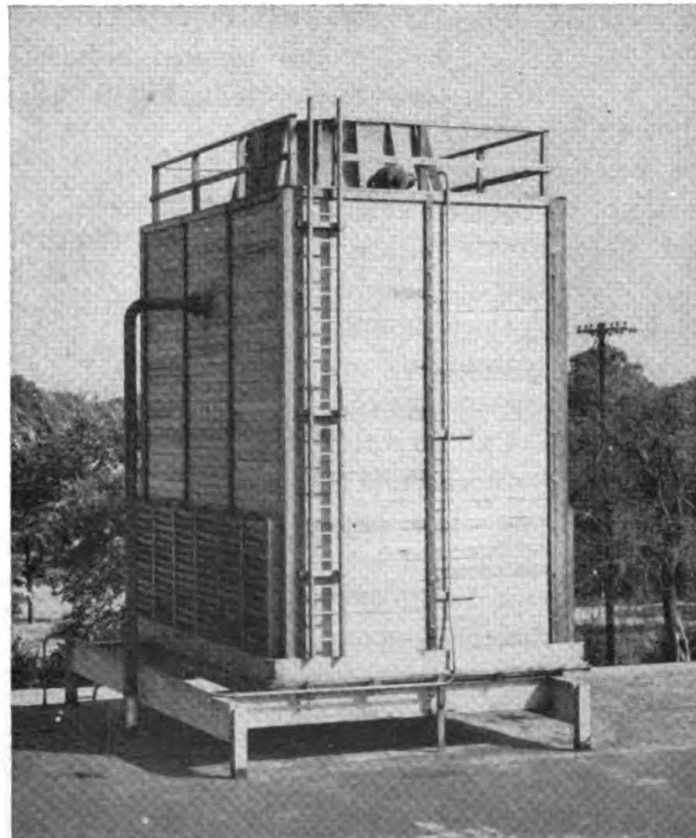


Fig. 11.12. Vertical induced-draft cooling tower. Courtesy The Marley Co., Inc.

velocities of one-half the average or less and located with the longest side perpendicular to the prevailing summer winds.

Forced-draft mechanical towers are made of wood, metal, or masonry.

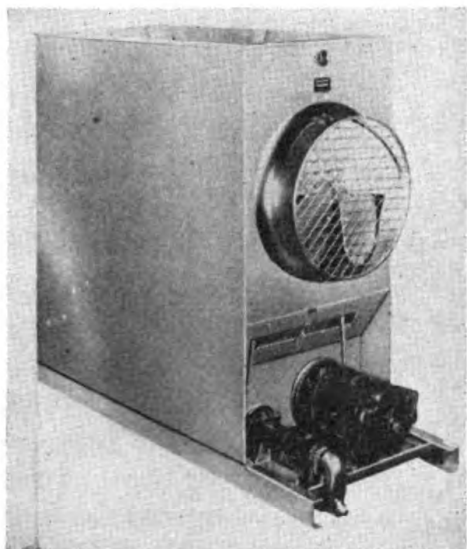


Fig. 11.13. Small horizontal induced-draft cooling tower. Courtesy The Marley Co., Inc.

The inside may be a wood lattice-work with water being discharged over the top, or the inside may consist of numerous spray nozzles. Openings are provided on the side near the bottom, and propeller fans blow air upward through the tower. Increasing the height, area, or air quantity improves effectiveness of the tower. Air velocities from 250 to 450 fpm over the gross area are used.¹²

Induced-draft towers are like forced-draft towers except that the fans are located at the top instead of at the bottom and draw the air upward through the tower. A conventional vertical induced-draft tower is shown in Fig. 11.12 and a small

horizontal induced-draft tower in Fig. 11.13.

Cooling-tower design is a specialized subject that will not be covered in this text. However, approximate data are often desired before specific data can be obtained from a cooling-tower manufacturer. If certain assumptions are made concerning entering and leaving air and water, the ratio of air to water quantity can be calculated by means of a heat-balance equation. The heat energy entering must equal the heat energy leaving. Maximum design conditions can be used for the entering air, and the leaving air will be from 90 to 100 per cent saturated. The heat-balance equation with reference to Fig. 11.14 per pound of dry air is

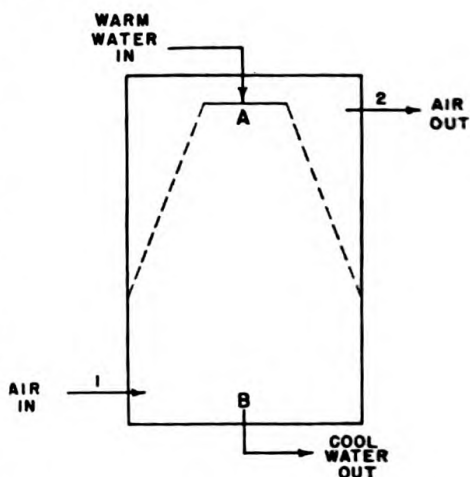


Fig. 11.14. Diagrammatic cooling tower.

$$h_{a1} + w_{H1}h_{v1} + w_w h_{fA} = h_{a2} + w_{H2}h_{v2} + w_w h_{fB} \quad (11.6)$$

¹² The preceding information on cooling towers is from *Heating-Ventilating Air Conditioning Guide 1948*. New York: American Society of Heating and Ventilating Engineers, Chapter 37.

where h_a = enthalpy of dry air

w_H = humidity ratio of air

h_v = enthalpy of moisture in air

w_w = weight of water per pound dry air

h_f = enthalpy of water.

An empirical formula sometimes used to determine the approximate leaving water temperature is

$$t_B = \frac{t_{db1} + 2t_{wb1} + t_A}{4} \quad (11.7)$$

The cooling effectiveness of a cooling tower is determined from the equation

$$\frac{t_R - t_A}{t_B - t_{wb1}} \times 100 = \text{per cent effectiveness} \quad (11.8)$$

EXAMPLE 11.1. Determine the approximate leaving water temperature from a cooling tower if the entering water is 96 F. Atmospheric conditions are 95 F dry bulb and 78 F wet bulb.

SOLUTION:

From equation 11.7,

$$t_B = \frac{95 + 156 + 96}{4} = 87 \text{ F approx.}$$

The cooling effectiveness in this case would be $\frac{96 - 87}{96 - 78} = 50$ per cent, which is a minimum value. Good tower design would increase this to over 60 per cent.

EXAMPLE 11.2. Determine the approximate amount of air to be handled and the quantity of make-up water required by a cooling tower that is to cool 200 gpm from 96 to 87 F. Atmospheric conditions are 95 F dry bulb and 78 F wet bulb. Assume that the air leaves the tower 90 per cent saturated at 90 F.

SOLUTION:

$h_{a1} + w_{H1}h_{v1}$ = enthalpy of entering-air mixture and from the psychrometric chart = $41.55 - 0.18 = 41.37$ Btu per pound.

$$w_{wA}h_{fA} = 64w_{wA}$$

At 90 F dry bulb, 90% relative humidity,

$$h_{a2} + w_{H2}h_{v2} = 52.5$$

and

$$w_{wB}h_{fB} = 55w_{wB}$$

Then

$$41.37 + 64w_{wA} = 52.5 + 55w_{wB}$$

But

$$\begin{aligned} w_{wB} &= w_{wA} - (w_{H2} - w_{H1}) \\ &= w_{wA} - (0.0280 - 0.0168) = w_{wA} - 0.0112 \end{aligned}$$

Therefore

$$41.37 + 64w_{wA} = 52.5 + 55w_{wA} - 55 \times 0.0112$$

or

$$w_{wA} = \frac{11.13 - 0.62}{9} = 1.17 \text{ lb water per lb dry air}$$

$$200 \text{ gpm would require } \frac{200 \times 8.33 \times 14.3}{1.17} = 20,300 \text{ cfm air}$$

$$\begin{aligned}\text{Evaporated water} &= (0.0280 - 0.0168) \times \frac{20,300}{14.3} \\ &= 15.9 \text{ lb per min}\end{aligned}$$

Make-up water = evaporated water plus some entrainment losses

Roughly, the make-up water for cooling towers is about 1 gpm per 1000 gal circulated for each degree of cooling.

11.9. Noncondensable Gases. Noncondensable gases are a mixture of nitrogen, oxygen, hydrogen, chlorine, and hydrocarbons. They may be present in the refrigerant drum used for charging the system, may be drawn into the system through leaks when part of the system is under a vacuum, may be in the system because of incomplete evacuation before starting, or may result from decomposition of the lubricant due to high discharge temperatures or from chemical reactions within the system. Their presence reduces the operating efficiency of the system by increasing the condensing pressure. They tend to form a film over parts of the condensing surface or to accumulate in the more quiet spaces in the condenser, reducing the heat-transfer coefficient or rendering part of the surface ineffective. Increased condensing pressure requires increased power requirements and reduces capacity.

The presence of noncondensable gases may be substantiated by allowing the condenser water to flow after shutting down the rest of the system. After sufficient time the refrigerant temperature should be the same as the water temperature. If the condenser pressure is then higher than the saturation pressure for the refrigerant temperature, noncondensable gases are present. A temperature difference of 10 F between the actual temperature and the saturation temperature of this condenser pressure may result in 20 to 25 F increase in the normal operating condenser temperature.

Purging may eliminate most of the gases. It is recommended that purging connections be located where these gases might collect. On horizontal condensers where the vapor enters at the top center, the top of the extreme ends of the condenser are suggested locations for purge connections. Special purging devices are also available.

11.10. Evaporator Theory. The design of an evaporator should provide for both effective boiling of the refrigerant with a minimum pressure drop and for efficient removal of heat from the medium being cooled. Heat is transferred to the outside surface mainly by convection when fluids are cooled and by conduction when solids are cooled, although the radiation factor also becomes very significant in some cases of free convection. This heat is conducted through the metal and is then transferred from the inside surface to the boiling refrigerant. The theory of refrigerant evaporators is rather complicated and involved, and little information on the correlation between theory and test values has been published.

In general, a rough, easily wetted surface produces higher boiling coefficients than smooth, oil-covered, or hard-to-wet surfaces. The coefficient increases as the temperature difference increases up to a certain point, depending upon the fluid and the surface, and then decreases.

The amount of heat transferred is equal to the rate of flow of the cooled fluid times its specific heat times its temperature drop. This amount is also equal to the rate of refrigerant flow times the difference between its leaving and entering enthalpy values.

11.11. Evaporator Design. Various metals are used for evaporators, depending upon the refrigerant to be used and the evaporator application; but iron, steel, and copper predominate. The actual design features of

TABLE 11.2
RETURN BEND TABLE¹³

<i>Size of Tubing or Pipe, in.</i>	<i>Equiv. ft of Tubing or Pipe per Return Bend</i>
$\frac{3}{8}$	1.25
$\frac{1}{2}$	1.50
$\frac{5}{8}$	1.50
$\frac{3}{4}$	1.75
$\frac{7}{8}$	1.75
1	2.50
$1\frac{1}{8}$	2.50
$1\frac{1}{4}$	3.00
$1\frac{1}{2}$	4.00
2	7.00

an evaporator depend upon the proposed location of the unit and whether the substance that is to be cooled directly is gaseous, liquid, or solid. In any event, the refrigerant either boils as it flows through a pipe, tube, or other type of space, sized so that liquid is continually wetting all the inside surface, or it boils in a shell around submerged tubes through which the fluid to be cooled is flowing.

When the refrigerant flows through a tube, an increased velocity theoretically gives greater heat transfer; but an increased velocity also causes greater pressure drop, which lowers the rate of refrigerant flow. Hence there are economical limits to the length and the diameter of tubing to use for given conditions. It is recommended that refrigerant tube-type evaporators for a given load have individual circuits of equal pressure drop that approach the maximum length for the diameter used, as given in Table 11.2, 11.3, and 11.4.

One of the basic equations for heat transfer is $q = UA\Delta t$. Applied to the outside surface of an evaporator, the equation becomes

$$q = f_o A_o (t - t_o) \quad \text{Btu per hr} \quad (11.9)$$

If the temperature t of the medium to be cooled is constant, if the outside

¹³ From *Refrigerating Data Book*, 5th ed. New York: American Society of Refrigerating Engineers, 1943, p. 391.

TABLE 11.3
MAXIMUM LOAD, BTU PER HOUR IN FREON-12 EVAPORATORS FOR EACH CIRCUIT¹⁴

Equivalent Length of Pipe or Tube per Circuit, ft	O.D. of Tube, in.						
	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	1	$1\frac{1}{8}$
40	1,130						
50	1,010						
60	920	2,400					
70	850	2,200	4,300				
80	795	2,050	4,000				
90	755	1,950	3,800				
100	720	1,850	3,550	5,800	8,400		
110	685	1,770	3,410	5,550	8,000		
120	655	1,690	3,270	5,300	7,700		
130	625	1,620	3,120	5,100	7,400		
140	605	1,550	3,010	4,900	7,120		
150	585	1,500	2,900	4,750	6,850	11,800	13,100
200	510	1,295	2,500	4,100	5,950	10,250	11,400
250	455	1,150	2,225	3,650	5,300	9,200	10,100
300	415	1,050	2,030	3,320	4,900	8,350	9,200
350	385	965	1,900	3,075	4,500	7,700	8,500
400	360	910	1,770	2,860	4,250	7,250	7,950
450	340	850	1,670	2,700	4,000	6,850	7,500
500	320	810	1,550	2,550	3,700	6,400	7,100
550	310	770	1,490	2,430	3,550	6,100	6,800
600	295	740	1,430	2,320	3,400	5,900	6,500
650	285	705	1,370	2,220	3,300	5,700	6,250
700	272	680	1,320	2,150	3,200	5,450	6,000
750	263	650	1,270	2,080	3,100	5,300	5,800
800	225	630	1,230	2,015	2,990	5,100	5,620
850	610	1,200	1,950	2,900	4,990	5,450
900	595	1,160	1,905	2,800	4,850	5,300
950	580	1,135	1,850	2,750	4,725	5,150
1,000	570	1,120	1,820	2,700	4,600	5,050
1,100	1,070	1,730	2,550	4,400	4,850
1,200	1,015	1,650	2,450	4,200	4,620
1,300	970	1,580	2,350	4,030	4,450
1,400	940	1,525	2,260	3,900	4,300
1,500	905	1,480	2,190	3,750	4,150
1,600	1,430	2,110	3,625	4,000
1,700	1,380	2,060	3,525	3,900
1,800	2,000	3,450	3,780
1,900	1,960	3,350	3,680
2,000	1,900	3,260	3,580

¹⁴ From *Refrigerating Data Book*, 5th ed. New York: American Society of Refrigerating Engineers, 1943, p. 390.

TABLE 11.4
MULTIPLIER TABLE FOR TABLE 11.2¹⁵

Air to Refrig. Temp. Diff.	Evaporator Temperature										
	-20	-15	-10	-5	0	5	10	15	20	25	30
10	.69	.77	.86	.96	1.06	1.18	1.31	1.45	1.59	1.74	1.90
15	.86	.97	1.08	1.20	1.32	1.46	1.61	1.78	1.95	2.13	2.32
20	1.00	1.12	1.23	1.37	1.51	1.68	1.87	2.06	2.27	2.49	2.71
25	1.14	1.28	1.42	1.57	1.73	1.90	2.10	2.31	2.55	2.81	3.08

surface film coefficient f_o is constant, and if the outside surface area A_o can be increased considerably without increasing the average surface temperature t_s too much, the rate of heat transfer will be increased. By using extended surface or fins on the outside of the tube, this increase can be obtained without changing the refrigerant flow. Therefore, as a matter of economy, refrigerant evaporators should be of the extended-surface or finned-tube type wherever practical. In order to keep the average surface temperature down, a good bond between the fin and tube is essential. Integral fins formed out of the tube itself are best in this respect and give the best heat-transfer rate per pound of metal used (see Fig. 11.8). Nevertheless, most air-cooling coils are of the bonded-fin type similar to the sections shown in Figs. 11.15 and 11.16.

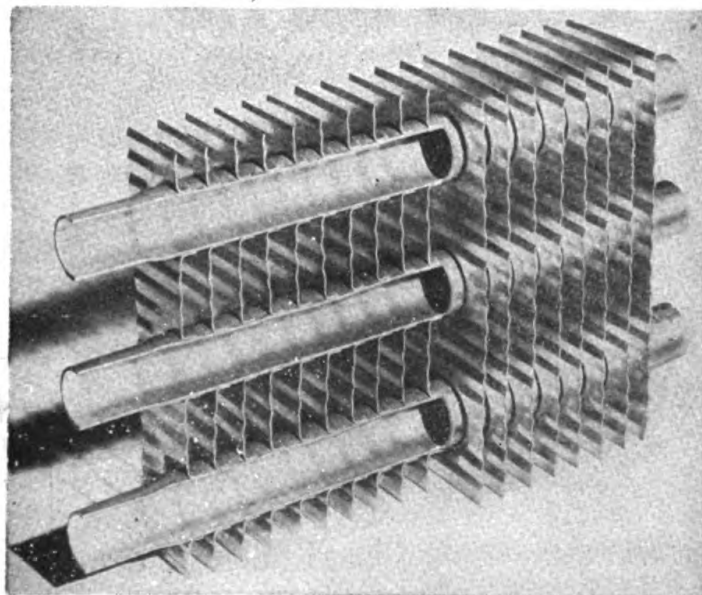


Fig. 11.15. Continuous-plate fin-coil section. Courtesy McQuay, Inc.

¹⁵ From *Refrigerating Data Book*, 5th ed. New York: American Society of Refrigerating Engineers, 1943, p. 391.

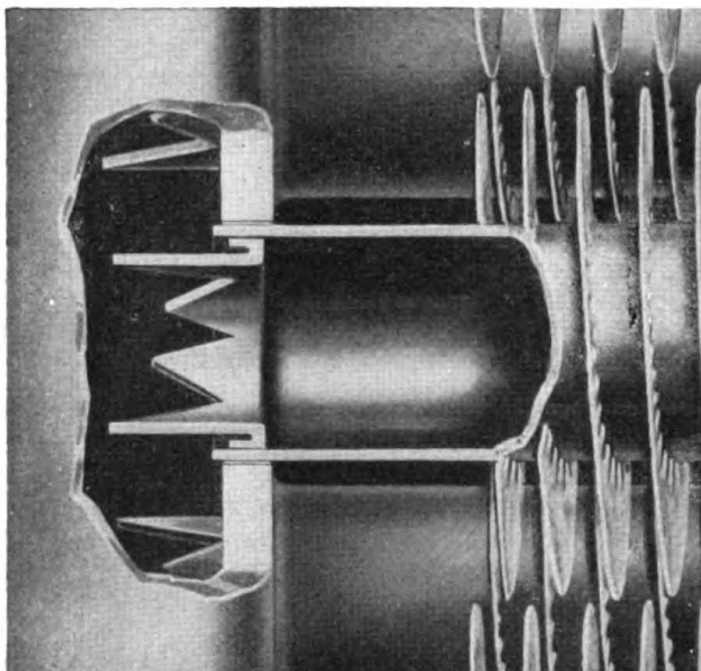


Fig. 11.16. Spiral fin-coil section. Courtesy Aerofin Corp.

EXAMPLE 11.3. Assume a 1-ton load in the nature of a brine to be cooled by 200 ft of submerged $\frac{3}{4}$ -in.-O.D. copper tube. The average brine temperature is 15 F and the refrigerant temperature is 0 F. How many equal parallel refrigerant circuits should be used if there are 40 return bends in the tubing?

SOLUTION:

$$\begin{array}{rcl} 1.75 \text{ ft per bend} \times 40 & = & 70 \\ \text{Tubing length} & = & 200 \\ & & \hline & & 270 \text{ total equivalent feet} \end{array}$$

$$\frac{12,000 \text{ Btu/hr}}{1.32} = 9,080 \text{ Btu per hr for}$$

Table 11.2

(factor 1.32 is from Table 11.4)
9,080 Btu is too great for 270 ft

Using two circuits gives 4,540 Btu per hr for 135 ft, which is satisfactory.

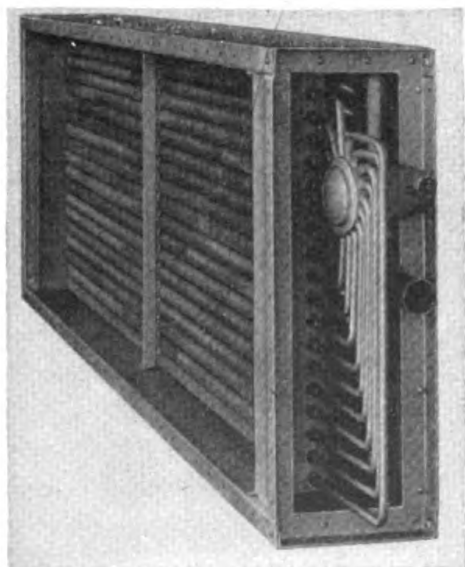


Fig. 11.17. Direct-expansion blast coil with refrigerant-distributing header. Courtesy Aerofin Corp.

ing air and other gases by forced convection. Figure 11.17 shows a typical blast coil with a refrigerant distributing header. A blast coil is placed in a duct or in an assembled unit, and the air is forced or drawn

11.12. Blast Coils and Unit Coolers. These are the terms given to refrigerant evaporators used for cooling

across it and discharged directly into the space or distributed through a system of ductwork to the rooms to be cooled. Unit coolers are very much like unit heaters, consisting of a coil, propeller-type fan, and the necessary controls, and are suspended in the space to be cooled. They are used in refrigerators and coolers of all types except the small domestic

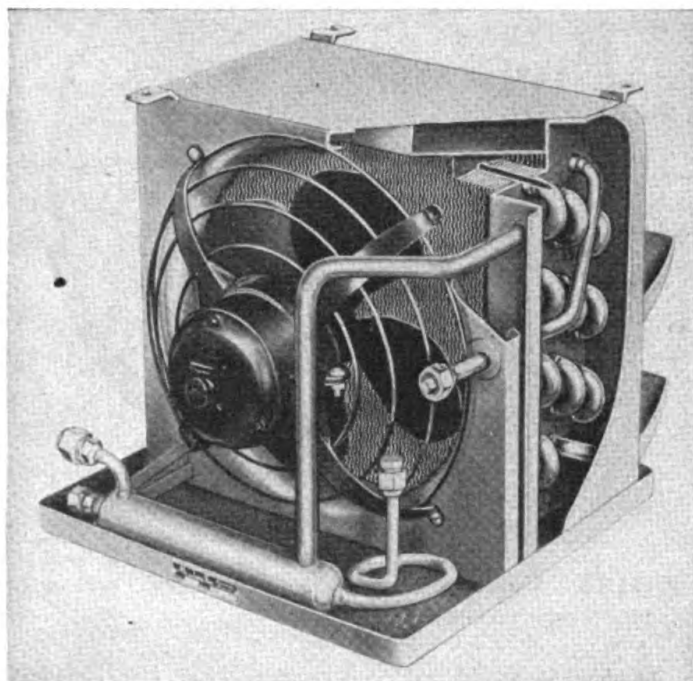


Fig. 11.18. Small unit cooler with liquid-suction heat exchanger. Courtesy McQuay, Inc.

boxes. When refrigerant temperatures below 32 F are required, some means of defrosting must be employed. Coils having only two to three fins per inch are recommended then instead of those having seven or eight per inch, which are used when there is to be no frost formation. Where extreme frosting occurs, it is often better to use plain tubes or prime surface evaporators, or possibly to provide some means for reducing the extreme frosting, such as a vestibule with a cooling coil in it.

Space temperatures around 0 F are less difficult to maintain than those around 30 F, since the low-temperature frost is less sticky and more snow-like, and easily blown off the surfaces.

Forced-air coolers are more efficient than gravity circulation units; less cooling surface is required and higher evaporator pressures can be used, thus saving on compressor power input. The evaporator temperature must be higher than for gravity coils if exposed food products are in the space, in order to prevent excessive dehydration. Small unit coolers have replaced many of the old ceiling-mounted or wall-mounted iron-pipe gravity coils like those shown in Figs. 1.1 and 1.5.

Air velocities from 300 to 600 fpm across the coil face are used; generally the range is between 400 and 500 fpm.

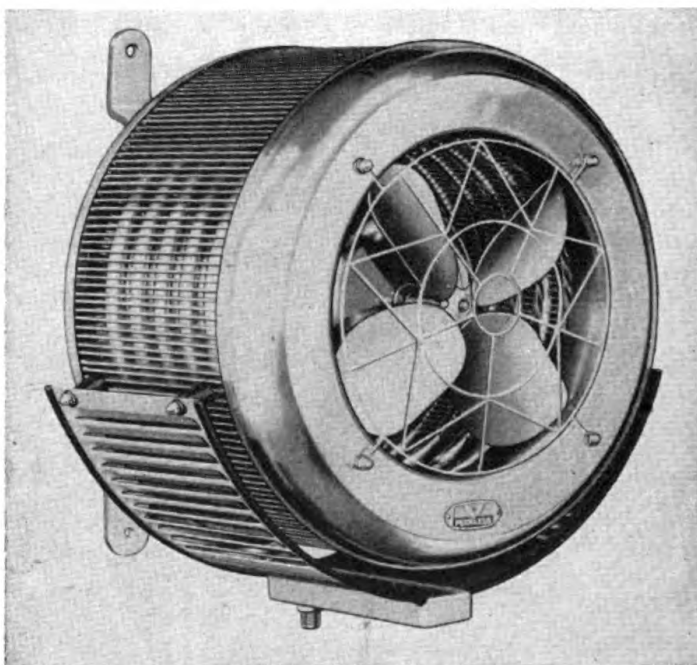


Fig. 11.19. Circular unit cooler. Courtesy Peerless of America, Inc.

Various types of unit coolers are shown in Figs. 11.18, 11.19, and 11.20.

Note the liquid-suction heat-exchanger and expansion-valve connections in Fig. 11.18.

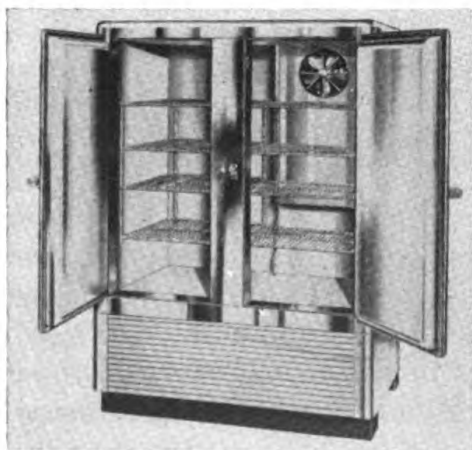


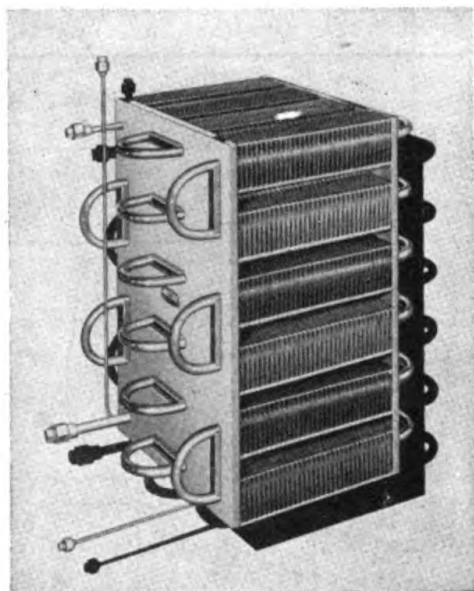
Fig. 11.20. 25 cu ft reach-in refrigerator with a unit cooler. Courtesy Koch Refrigerators.

11.13. Gravity Coils. Coils of scale-free iron pipe were used extensively for cooling air by gravity circulation before extended-surface evaporators were developed. Pipes of 1 to 2 in. diameter were generally formed into coils for mounting on the ceiling or wall or were used as shelves. For such coils, heat-transfer values vary from 1.6 to 2.3 Btu per (hr)(sq ft) (F), increasing as the temperature difference increases and also as the space temperature is increased with

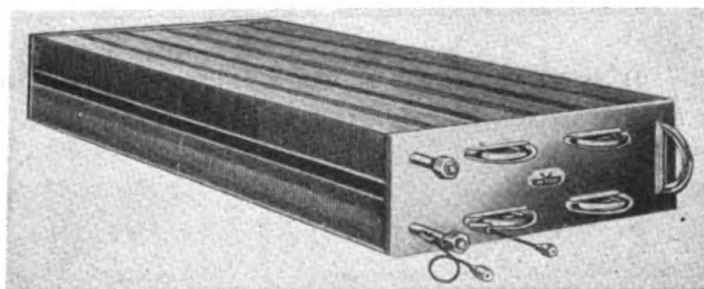
the same temperature difference. Capacities for 1-in. pipe coil are given in Table 11.5.

As methyl chloride and the Freons came into use, finned copper coils were developed for gravity air cooling. Various types of gravity coils are

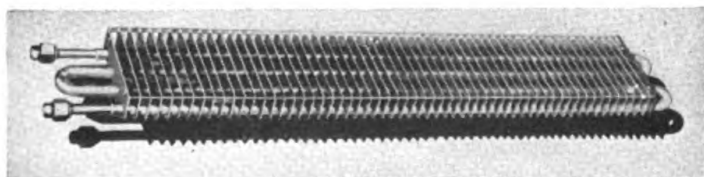
shown in Fig. 11.21, and a typical gravity air cooler with double condensate collectors is shown in Fig. 11.22. These gravity assemblies are used primarily in commercial refrigeration fixtures where it is not feasible to use unit coolers because of space limitations or because of objections to fans or motors. Suggested locations of coils and baffles are shown in Fig. 11.23. Baffles are often placed near the coils to improve the air circulation, and the location of the coil as well as the temperatures involved affect the heat-transfer rates. Previously unpublished test results of Priester indicate that when no baffles were used, locating a finned gravity coil 3 in. below the ceiling increased its capacity $33\frac{1}{3}$ per cent over that at a location 1 in. below the ceiling. However, when the coil was located 7 in. from the ceil-



a



b



c

Fig. 11.21. Gravity air-cooling coils. Courtesy Peerless of America, Inc.

ing, the capacity was 50 per cent over that at 1 in., or only $12\frac{1}{2}$ per cent over that at 3 in.

Ratings for finned coils are difficult to predict, so that capacities published by reliable manufacturers must be used. In general, heat-transfer values from 0.4 to 2.0 Btu per (hr)(sq ft)(F) are recommended, decreas-

ing as the fin space is decreased and as the number of rows in height is increased.

TABLE 11.5
PIPE COIL CAPACITIES, GRAVITY AIR CIRCULATION¹⁶
Btu/(hr)(lin. ft) for 1-in. standard or extra-heavy pipe

Refrig. Temp., F	Fixture Temperature, F				
	-20	0	20	30	40
25	11.1
20	16.2
15	9.3	21.0
10	12.7	25.8
5	7.5	16.3	30.4
0	11.0	19.9	34.9
-5	14.6	23.5	
-10	18.4	27.5	
-15	...	7.5	21.6		
-20	...	11.0	24.8		
-25	...	14.6			
-30	...	18.3			
-35	7.3	21.5			

Additional surface over the amount computed to handle the load should be used for direct-expansion gravity evaporators, to allow for some

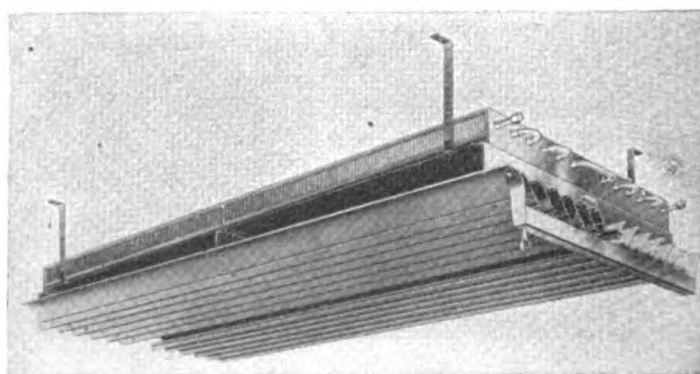
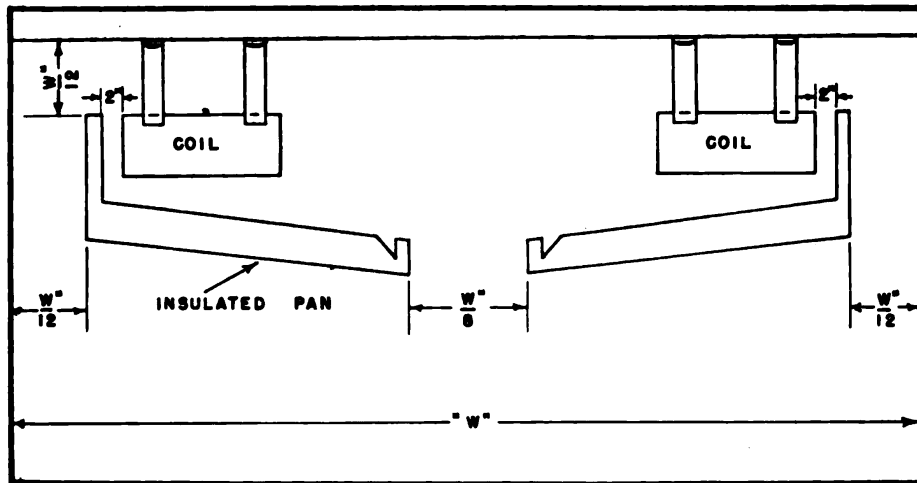


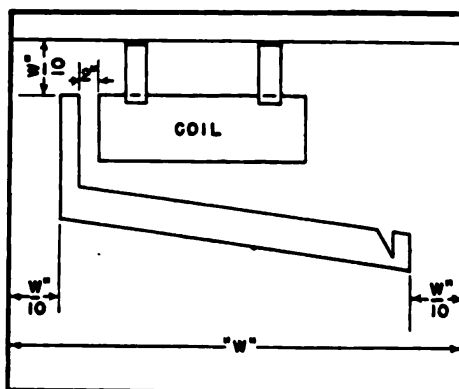
Fig. 11.22. Gravity air-cooling unit. Courtesy McQuay, Inc.

superheating of the vapor. This additional surface is called the drier surface. Some manufacturers recommend 25 per cent additional surface for "drying" unless a liquid suction heat exchanger is used.

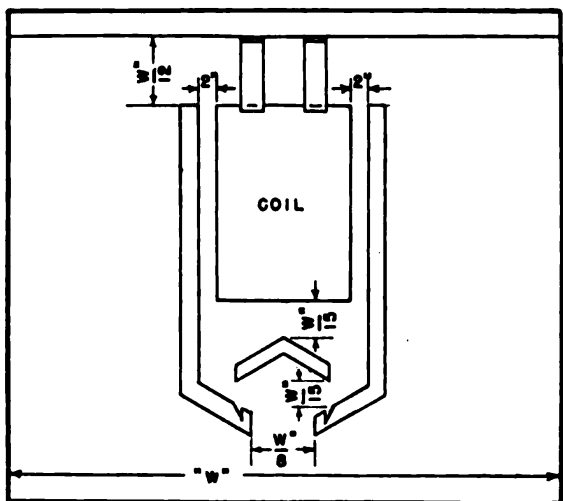
¹⁶ *Refrigerating Data Book*. 5th ed. New York: American Society of Refrigerating Engineers, 1943, p. 389.



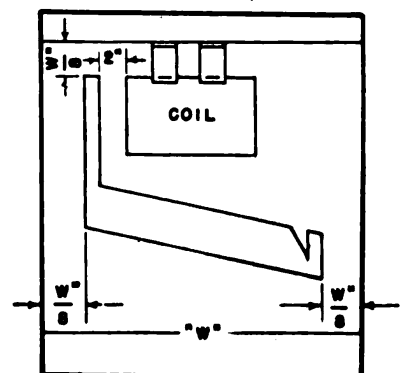
(a) Walk-in refrigerator, " W " is over 8 ft.



(b) Walk-in refrigerator, " W " is under 8 ft.



(c) Reach-in refrigerator.



(d) Reach-in refrigerator.

Figs. 11.23, (a), (b), (c), (d).

Suggested gravity-coil and baffle locations in refrigerators. Courtesy Peerless of America, Inc.

11.14. Plate Evaporators. The plate type of evaporator is replacing other types of gravity coils to a great extent. Refrigerated plates similar to those shown in Fig. 11.24 are used in many home freezers, display cases, refrigerated trucks, beverage coolers, ice-cream cabinets, locker plants,

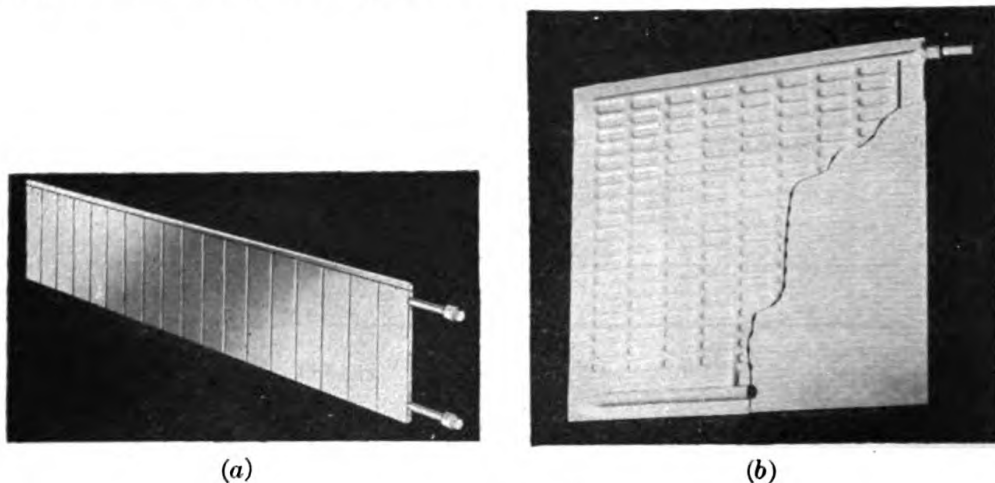


Fig. 11.24. Plate-type evaporators. (a) Courtesy Peerless of America, Inc. (b) Courtesy The Yoder Co.

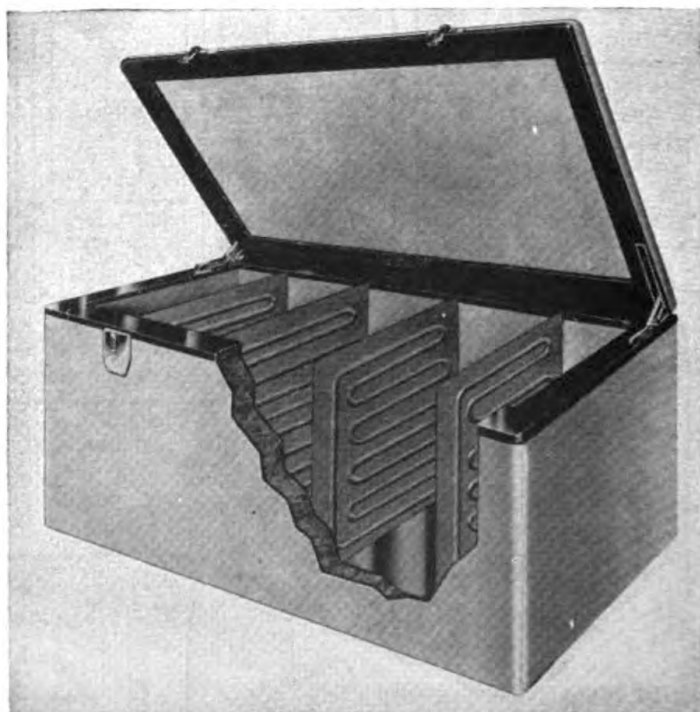


Fig. 11.25. Plate evaporators in a freezer cabinet. Courtesy Kold-Hold Mfg. Co.

and so on. Manufacturers of plates recommend using heat-transfer coefficients varying from 1.5 to 2.5 Btu per (hr)(sq ft)(F), using the area of both sides of the plates. When plates or coils are used as shelves with the product to be chilled resting directly on them, the heat-transfer

coefficient is increased 15 to 25 per cent. One method of applying plates to a freezer unit is indicated in Fig. 11.25.

11.15. Liquid Chillers. Various types of refrigerant evaporators are used for chilling liquids. The simplest consists of submerged pipe or tube coils usually placed around the sides of a tank. A heat-transfer rate in nonagitated water of 30 Btu per (hr)(sq ft)(F) is recommended for direct expansion coils and a rate of 40 for flooded coils. Values of 20 may be used for direct expansion coils and 25 for flooded coils in brine

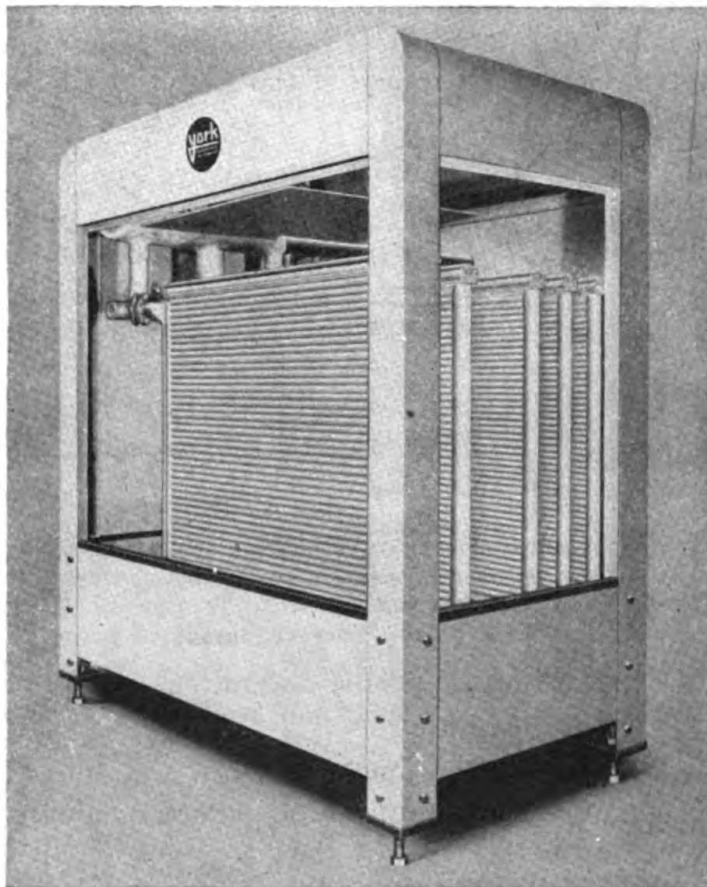


Fig. 11.26. Baudelot-type liquid cooler. Courtesy York Corp., York, Penna.

baths above 0 F. These values decrease somewhat with a decrease in liquid temperature.

Shell-and-coil coolers and shell-and-tube coolers, very much like the condensers of the same name, are also used, particularly for chilling water or brine, which is then pumped to air cooling coils in various parts of a building. Automatic protection against freezing should be provided when water is to be chilled to a temperature below 40 F.

Double-pipe chillers similar to condensers have been used in dairy, beverage, and oil plants. Baudelot-type coolers similar to Fig. 11.26 are generally used in dairy and beverage plants, where liquids must be cooled

is at the room conditions, and AB is the line representing the room sensible-heat ratio. The required leaving-coil condition is at point C where XY and AB intersect and is maintained by controlling the coil-refrigerant temperature.

In order to obtain the desired room conditions, each pound of saturated supply air will pick up sensible heat equal to its specific heat times the difference between the room dry-bulb temperature and the apparatus dew-point temperature. Each pound of saturated air will also pick up total heat equal to the difference between the enthalpy at the room conditions and the enthalpy at the apparatus dew point. The ratio of the sensible heat to be picked up to the total heat to be picked up is also the *room sensible-heat ratio*. Expressed in equation form,

$$\frac{\text{Room sensible heat}}{\text{Room total heat}} = \text{room sensible-heat ratio} \\ = \frac{0.24 (\text{room dry bulb} - \text{apparatus dew point})}{h_{\text{room condition}} - h_{\text{app. dew point}}} \quad (11.10)$$

After the sensible-heat ratio has been determined, the necessary apparatus dew-point temperature is found by a cut-and-try procedure with equation 11.10 or from a table previously made from calculated values¹⁷ by means of equation 11.10. The cfm of air to be cooled by the coil can then be calculated from the equation

$$\text{cfm} = \frac{\text{room sensible heat}}{1.08 (1 - \text{by-pass factor}) (\text{room d.b.} - \text{apparatus dew point})} \quad (11.11)$$

Should equations 11.10 and 11.11 indicate an apparatus dew point and air quantity that for some reason are not desired or feasible, it is suggested that a change in the room conditions be tried, using the comfort zone of the effective temperature chart (Fig. 10.1). New conditions of about equal comfort may be found that will change the sensible and latent loads slightly, and, when used in equations 11.10 and 11.11 with the revised sensible-heat factor, will indicate a more desirable apparatus dew point and coil air quantity. In certain critical industrial or commercial applications involving product conditioning, the room conditions cannot be altered, and artificial sensible heat is sometimes added to the load by means of reheat coils in supply air ducts, electric heat sources in the space, and so on. Since this additional heat raises the apparatus dew point and the compressor suction temperature required, it may not increase the total power and operating costs even though the total refrigeration load is increased.

¹⁷ Many engineers have used values of sigma heat content or total heat content in place of enthalpy values, since they have been easier to determine and the resulting accuracy is sufficient for all but the most special installations.

The coil refrigerant temperature required is selected from the rating tables of coil performance based on the grand total heat load, including the outside air load. If no by-pass factors are given, the coil manufacturer generally furnishes other data that can be used for selecting coils. (The average four-row air-conditioning coil of seven fins per inch has a by-pass factor of 0.2.)

Most air-conditioning applications have coils three to eight rows deep in the direction of air flow. In general, more rows of coil result in less air and higher refrigerant-temperature requirements. Also, the lower the room sensible-heat ratio, the greater the number of rows of coil usually needed to give the most economical balance of total equipment required.

When, due to lack of reliable information on the amount of latent heat from the product in the space, it is not possible to obtain the sensible-heat ratio, the apparatus dew point or coil temperature must be based on a selected difference between the room temperature and coil temperature. Maximum values of this temperature difference and maximum room air velocities for various products are given in Table 11.6.

Gravity coils and plates are selected from manufacturers' tables, based on the total room load and the desired difference of room temperature minus refrigerant temperature. The refrigerant temperature is selected at a given temperature difference below the desired room temperature. This temperature difference depends upon the relative humidity desired in the room. The higher the relative humidity wanted, the smaller the temperature difference should be. Relative humidity is important only where exposed products are to be kept in the room. The following general temperature differences between refrigerant and room dry bulb are recommended on the basis of experience:

Fresh or cut meats, poultry, cheese, eggs, fresh fruits, fresh vegetables, and so forth	} . . . 15 to 20 F
Dried and smoked meats, dried fruits, dried vegetables, iced fish, and products in wooden containers	} . . . 20 to 25 F
Canned, bottled, or moistureproof sealed con- tainers or products	} . . . Any temperature difference.

An economic balance of equipment cost and operating cost determines the optimum refrigerant temperature for water chillers. A low refrigerant temperature requires less cooling surface than a high one but greater compressor capacity and power requirements.

EXAMPLE 11.4. Select the apparatus dew point and cfm required for the load shown in Fig. 10.8.

TABLE 11.6
RECOMMENDED MAXIMUM TEMPERATURE DIFFERENCE AND AIR VELOCITY
FOR VARIOUS COMMODITIES WITH FORCED CONVECTION COOLING¹⁸

Commodity	Maximum Temperature Difference, F	Maximum Air Velocity, ft/min
Beef (quarters)	12	40
Beer (wooden kegs)	12	40
Berries	12	40
Butter	12	40
Cheese	12	25
Citrus fruits	12	40
Cut meats	12	25
Dried vegetables	16	60
Eggs (crated)	12	40
Fish (iced)	16	60
Flowers (cut)	12	25
Fruits:		
Dried	16	60
Fresh	12	40
Lamb (halves)	12	40
Lard	20	100
Malt	20	100
Maple sugar	20	100
Maple syrup	20	100
Meats:		
Dried	16	60
Hams (fresh)	12	40
Loins (fresh)	12	40
Sausage (bulk)	12	25
Smoked	16	60
Nuts (dried)	16	60
Onions	16	60
Oysters	16	60
Pork (halves)	12	40
Poultry (fresh)	12	40
Sausages	12	40
Veal (halves)	12	25

¹⁸ From *Refrigerating Data Book*, 5th ed. New York: American Society of Refrigerating Engineers, 1943, p. 396.

Room sensible heat = 84,900 Btu per hr
 Room total heat = 92,800 Btu per hr
 Grand total heat = 105,600 Btu per hr
 Room conditions: 80 F d.b., 67 F w.b., 31.51 Btu/lb enthalpy

$$\text{Room sensible-heat ratio} = \frac{84,900}{92,800} = 0.92$$

SOLUTION:

Assuming 58 F apparatus dew point,

$$\begin{aligned} \frac{0.24(80 - 58)}{31.51 - 25.12} &= \frac{0.24 \times 22}{6.39} = 0.83 \text{ (too low)} \\ \text{For 59 F, } \frac{0.24(80 - 59)}{31.51 - 25.78} &= \frac{0.24 \times 21}{5.73} = 0.88 \text{ (too low)} \\ \text{For 60 F, } \frac{0.24(80 - 60)}{31.51 - 26.46} &= \frac{0.24 \times 20}{5.05} = 0.95 \text{ (too high)} \end{aligned}$$

If the room conditions were changed to 80 F dry bulb, 66 F wet bulb, the latent heat load would be changed slightly:

Outside air	375 cfm	$\times 44.2 \times 0.2 \times 0.67$	=	2,200 Btu per hr
People	25	$\times 180$	=	4,500
Gas burner			=	700
				<u>7,400</u>
		+10%		<u>700</u>
Room latent heat			=	8,100
Room sensible heat			=	84,900
Room total heat			=	<u>93,000 Btu per hr</u>
				$\frac{84,900}{93,000} = 0.91$ room sensible heat ratio

375 cfm	$\times 0.15 \times 0.8 \times 1.08$	=	8,900
375	$\times 44.2 \times 0.8 \times 0.67$	=	5,000
Grand total heat		=	106,900

Again trying 58 F apparatus dew point but 80 F d.b., 66 F w.b.

$$\frac{0.24(80 - 58)}{30.72 - 25.12} = \frac{0.24 \times 22}{5.60} = 0.94$$

Usually it is not economical to design applications for apparatus dew points much above 58 F. For this particular problem a 58 F design apparatus dew point is recommended. This would produce room conditions of 80 F dry bulb and 66 to 67 wet bulb, requiring less than 106,900 Btu per hour total cooling.

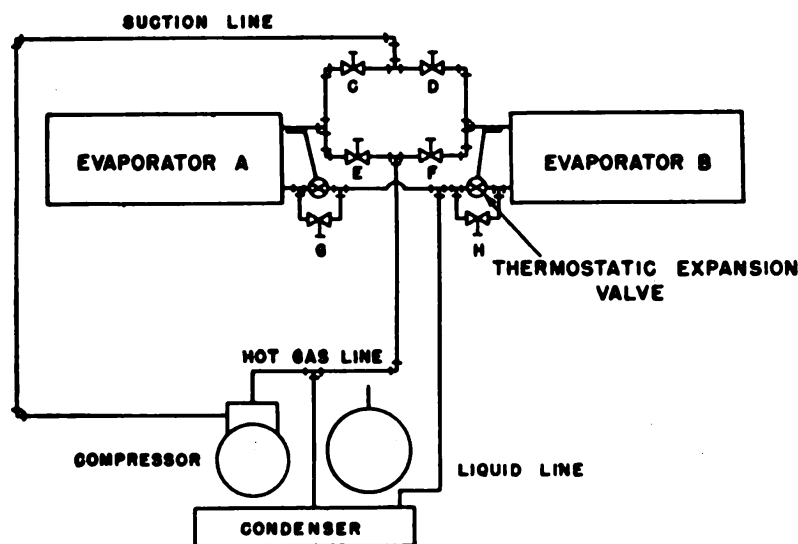
The coil-air quantity, using a four-row coil, is

$$\frac{84,900}{(1 - 0.2)(80 - 58)1.08} = 4500 \text{ cfm}$$

One manufacturer makes a factory-assembled unit using a four-row coil of 9 sq ft face area, which produces an air velocity of 500 fpm. This unit is a little smaller than those recommended for quiet applications but should be satisfactory for this industrial application.

11.17. Defrosting. Various methods of removing the frost accumulation on evaporators are used. When the space temperature is above 32 F, forced-air convectors can be defrosted by letting the fan run during the off-cycle periods of the compressor. The normal off-cycle period, however, is usually not long enough to defrost gravity convectors, and the compressor must be shut off manually.

In rooms held at temperatures below freezing, some artificial means of defrosting must be employed. When large pipe coils or plates are used, particularly on installations of older design, the frost is allowed to accu-



NORMAL OPERATION	- C & D OPEN, E, F, G & H CLOSED
DEFROSTING A	- D, E & G OPEN, C, F & H CLOSED
DEFROSTING B	- C, F & H OPEN, D, E & G CLOSED

Fig. 11.28. Hot-gas defrosting system.

multate for several weeks or months and is then scraped off manually. This is a disagreeable task. In the meantime the frost tends to insulate the coil, resulting in inefficient operation.

Hot-gas defrosting is used on many installations. The equipment then is piped and valved so that the operations of the condenser and evaporator can be interchanged. To defrost the regular evaporator, hot vapor from the compressor is discharged to it, melting the frost as the refrigerant condenses, and the regular condenser serves as an evaporator. Precautions must be taken to prevent trapping the oil and freezing the condenser. Figure 11.28 is a schematic diagram of a two-evaporator system.

When large factory-assembled unit coolers are used, intake and discharge ducts with diverter dampers to a space outside the cooler can be added. In order to defrost, the dampers are repositioned, cutting off air

circulation to the cooler and circulating air from and to the space outside the cooler.

For temperatures not too far below freezing, brine spray units serve very well. These units resemble the evaporative condenser, as shown in Fig. 11.29. The coil is a refrigerant evaporator, however, and sometimes is finned. A portion of the drain tank is partitioned off and filled

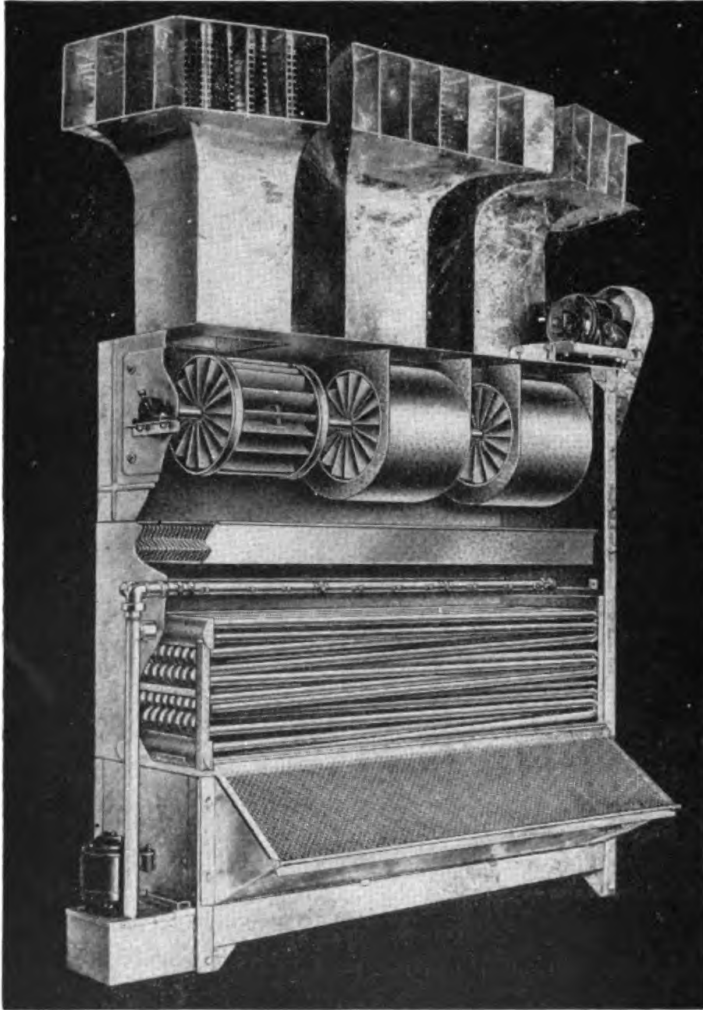


Fig. 11.29. Brine-spray unit cooler. Courtesy Carrier Corp.

with salt. The concentrated salt solution overflows into the main part of the drain tank and tends to maintain the spray solution at a constant concentration. Sodium brines have higher freezing temperatures than calcium brines. Calcium brines, however, should not be sprayed if exposed food products are in the cooled space.

A water spray used periodically over the coil (see diagram, Fig. 11.30) is one of most successful defrosting methods. It is becoming widely used, particularly for the higher temperatures. The by-pass connection is suggested to prevent the drain from freezing up. Other special defrosting

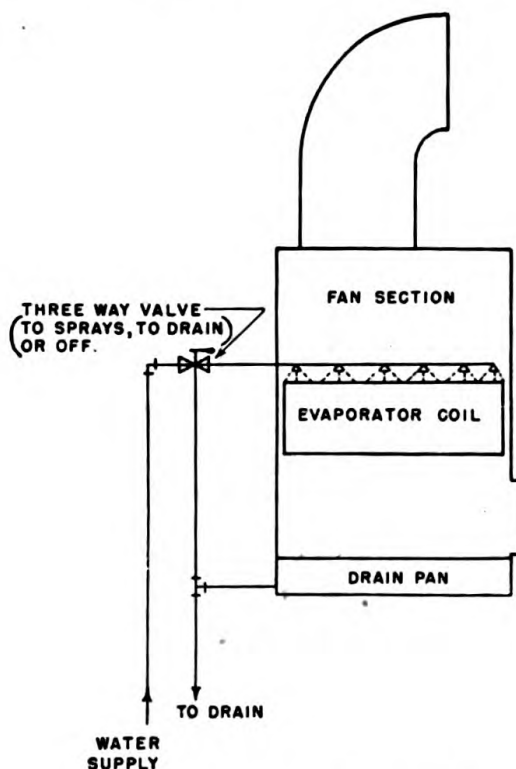


Fig. 11.30. Water-defrosting system.

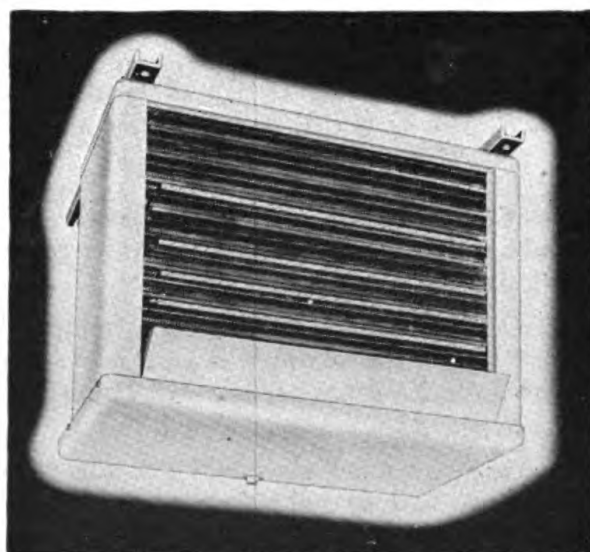


Fig. 11.31. Electric heater-defrosted unit cooler. Courtesy Bush Mfg. Co.

methods employ heat exchangers, electric heaters, and other devices, but the water spray is the simplest for many applications.

One company has introduced an automatic water-defrosting device that is connected to the fan motor of a unit cooler. When, because of additional coil resistance from frost accumulation, the current to the fan

motor increases to a predetermined value, the fan is stopped, the liquid-line solenoid valve is closed, and the water solenoid valve is opened. After a fixed interval the water solenoid is closed, the water is allowed to drain, and the cooling cycle is started again.

A unit cooler with self-contained electric resistance defrosting equipment is shown in Fig. 11.31. The electric elements are visible on the face of the unit.

For rooms at 0 F and colder, the installation of a separate entrance room or space is recommended, with a gravity coil located there to keep the temperature about 36 F. Most of the moisture from the entering air will collect on this coil, thus requiring less frequent defrosting of the coils in the main freezer space.

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PROBLEMS

11.1. How many feet of $1\frac{1}{4}$ - \times 2-in. double-pipe condenser are necessary for each pound of ammonia vapor at 30 F superheat when condensing at 80 F if 15 gpm of water entering at 70 F are available?

11.2. Determine the most economical condensing temperature for a 10-ton Freon-12 system at 40 F evaporator if water at 75 F is available at \$1 per 1000 cu ft and electric power costs 2.5 cents per kilowatt-hour. What would be the cost of operation in cents per hour?

11.3. (a) Approximately how much air would a cooling tower need to handle at 95 F dry bulb and 78 F wet bulb to cool 100 gpm from 100 F to 85 F? Assume that air leaves the tower at 95 F and 90 per cent saturated. (b) Calculate the rate of water evaporation.

11.4. Determine (a) room sensible-heat ratio, (b) apparatus dew-point temperature, and (c) cfm of air to be cooled if the coil by-pass factor is 0.2 for the following load figures: room sensible heat, 84,000 Btu per hour; room latent heat, 16,000 Btu per hour; room air conditions, 78 F and 55 per cent humidity.

11.5. How many equal and parallel F-12 circuits should be used for an 18,000 Btu per hour water-chilling unit made of $\frac{5}{8}$ -in.-O.D. tubing? The average water temperature is 45 F and the coil is at 35 F. There are forty-eight 90-deg bends.

11.6. How many feet of 1-in. gravity pipe coil are needed for a 0 F room if the cooling load is 3 tons and the coil temperature is -20 F?

11.7. Determine the dehumidified-air quantity and apparatus dew-point temperature required for a vegetable-storage room at 32 F and 90 per cent relative humidity if the room sensible-heat load is 18,700 Btu per hour and the total room load is 27,600 Btu per hour. The coil by-pass factor is 0.6. An additional 12 per cent air quantity should be provided to allow for frosting of the coil.

11.8. For 80 F dry bulb and 67 wet bulb the subtotals are calculated to be 394,000 Btu per hour room sensible heat and 330,800 Btu per hour room latent heat. (In addition miscellaneous items are 15 per cent of room sensible and 10 per cent of room latent subtotals.) Calculate the apparatus dew point and air quantity for a four-row coil (by-pass factor is 0.2). If there are 1000 people, of whom 500 are dancing half the time and the rest remain seated (see Table 10.12), and the room conditions are changed to 76 F dry bulb and 66 F wet bulb (occupants' sensible heat increases 20 per cent and latent heat decreases accordingly), determine the new apparatus dew point and dehumidified-air quantity, using an eight-row coil (by-pass factor is 0.0); 10,000 cfm of outside air is used. Outside conditions are 88 F dry bulb and 76 F wet bulb. Neglect infiltration and the difference in transmission gain for the two conditions.

11.9. Experimental data for a clean condenser tube indicate the following U values at the turbulent water velocities shown: 833 at 7.45 ft per sec, 500 at 3.15, 370 at 1.89, and 294 at 1.25. Tests on a used tube give 589 at 7.45, 400 at 3.15, 334 at 2.2, and 250 at 1.25. If the tube is copper with inside and outside diameters of 0.902 and 1.0 in., respectively, calculate the vapor-side coefficient.

11.10. Determine the air quantity required for a 32 F room at 90 per cent relative humidity if the room sensible heat is 18,700 Btu per hour and the room total heat is 27,600 Btu per hour. Assume 0.5 by-pass factor and add 33 per cent air because of frost formation on the coil.

CHAPTER 12

Refrigeration Piping

12.1. Purpose. Pipe or tubing is used to convey the refrigerant from one unit to the next in a refrigeration system. In a simple cycle the mass rate of refrigerant flowing from each unit is the same, but the properties of the refrigerant, particularly the phase and the density, differ. Therefore, in order to use the most economical pipe, from the point of view of heat transfer and friction as well as pipe cost, pipe of different sizes is recommended for the different parts of the system. Furthermore, oil is carried with the refrigerant leaving a reciprocating compressor, and means for returning it to the crankcase must be provided. Pipe of the proper size aids the return of oil. This must be considered, particularly if oil separators are not used. The pipe must be of a material that will not corrode or react with the refrigerant. Fittings and types of joints should be selected that are economical, easy to install, and remain leakproof.

In the selection of pipe sizes, both pressure drop and velocity must be considered. The pipe size should not be so small that the friction loss is excessive or objectionable noise is generated due to high velocity. Neither should the pipe size be so large that excessive piping cost and poor oil return result. The ideal velocity depends on the state of the fluid, the quantity of refrigerant to be circulated, and the pressure drop.

12.2. Materials. Copper tubing is generally used for Freon-11, Freon-12 and Freon-22. It has also been used for the other Freons, methyl chloride, and sulfur dioxide. There are two general classifications of copper tubing: (1) small-diameter dehydrated tubing and (2) so called "water" tubing classified as Types K, L, and M in ASTM Specification B88.¹

The first comes in soft temper and is dehydrated at the mill. Generally it is in coils of 50 to 100 ft, with the ends sealed to keep air and moisture out. It is acceptable for connections in and around the smaller sized units. The fittings and joints are usually of the flared type. The tube wall thickness is 0.035 in. and the most common sizes are given in Table 12.1.

Types K, L, and M tubing come in both soft and hard temper. The soft tubing is made in coils of 30 to 60 ft in sizes to 1½ in. O.D. All sizes

¹ Most of the information in this section is adapted from Crocker, S., *Piping Handbook*, 4th ed. New York: McGraw-Hill Book Company, Inc., 1945, Chapter XVI.

TABLE 12.1
SMALL-DIAMETER COPPER TUBING—0.035-IN. WALL THICKNESS²

Outside dia., in.	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$
Weight, lb per ft	0.038	0.065	0.092	0.118	0.145	0.172	0.198	0.251	0.305

in both hard and soft temper are available in 20-ft lengths. The outside diameter of each type is $\frac{1}{8}$ in. larger than the nominal or standard water-tube size. In refrigeration work the size is usually identified as the outside diameter; but to avoid confusion in specifying or ordering tubing, it should be clearly indicated whether the size given means nominal size or outside diameter. Dimensions of the three types are given in Table 12.2.

The weight of tubing to select depends upon the application. The ASA B9 Safety Code for Mechanical Refrigeration states that soft annealed copper tubing erected on the premises shall not be used in sizes over $\frac{3}{4}$ in.; that Type L tubing or heavier must be used for field assembly, and Type M or heavier for shop assembly.

Copper or any of its alloys must not be used with ammonia. Brass and other copper-alloy pipes or tubing are sometimes used with the other refrigerants but are generally confined to condenser or evaporator tubes.

The ASA B31 Standard Code for Pressure Piping, 1942,³ prohibits the use of aluminum-base alloys for Freon-12 and methyl chloride, of magnesium-base alloys for all Freons and methyl chloride, and of zinc for methyl chloride. Any materials may be used with carbon dioxide and also with sulfur dioxide, provided no moisture is present.

The weight and bulk of seamless-steel, welded-steel, wrought-iron, and cast-iron pipe, and the type of fittings they require, prevent their widespread use with the Freon refrigerants except in sizes above 5 in. These are the only materials used to any extent for ammonia, however. Dimensions of welded and seamless steel pipe are shown in Appendix Table A.21 for Schedule 40 (standard) and for Schedule 80 (extra strong) thicknesses. The ASA B9 Safety Code states that for ferrous pipe the refrigerant working pressure shall not exceed 250 psi in Schedule 40 pipe; Schedule 80 pipe shall be used for sizes smaller than $1\frac{1}{2}$ in. carrying liquid refrigerant; and pipe thinner than Schedule 40 shall not be threaded.

² From Crocker, S., *Piping Handbook*, 4th ed. New York: McGraw-Hill Book Company, Inc., 1945, p. 1245.

³ These codes are available from the American Standards Association, 70 East 45th Street, New York 17, N.Y.

TABLE 12.2
DIMENSIONS OF COPPER TUBING⁴

Nominal Size, in.	Outside Diameter, in.	Wall Thickness, in.		
		Type K	Type L	Type M
$\frac{1}{8}$.250	.032	.025	.025
$\frac{1}{4}$.375	.032	.030	.025
$\frac{3}{8}$.500	.049	.035	.025
$\frac{1}{2}$.625	.049	.040	.028
$\frac{5}{8}$.750	.049	.042	.030
$\frac{3}{4}$.875	.065	.045	.032
1	1.125	.065	.050	.035
$1\frac{1}{4}$	1.375	.065	.055	.042
$1\frac{1}{2}$	1.625	.072	.060	.049
2	2.125	.083	.070	.058
$2\frac{1}{2}$	2.625	.095	.080	.065
3	3.125	.109	.090	.072
$3\frac{1}{2}$	3.625	.120	.100	.083
4	4.125	.134	.110	.095
5	5.125	.160	.125	.109
6	6.125	.192	.140	.122
8	8.125	.271	.200	.170
10	10.125	.338	.250	.212
12	12.125	.405	.280	.254

The strength of a pipe is determined by its thickness. The Piping Code requires a thickness based on the equation

$$x = \frac{pd}{2S} + C \quad (12.1)$$

where x = minimum wall thickness, inches

p = maximum internal pressure plus allowance for brittleness at low temperatures (2 per cent per degree below 0 F for ordinary ferrous metals, none for copper), psi

d = outside pipe diameter, inches

S = allowable stress in material due to p (see Table 12.3), psi

C = allowance, in inches, for threading, mechanical strength, and corrosion, as follows:

⁴ From American Society for Testing Materials Standard Specification for Copper Water Tube, Serial B88-46, Table II, *A.S.T.M. Book of Standards, Part IB, Non-Ferrous Metals*, Philadelphia, 1946.

Threaded pipe, under $\frac{1}{2}$ in.	0.05 in.
Threaded pipe, $\frac{1}{2}$ in. and over.	0.065 in.
Grooved pipe.	Depth of grooves
Plain end ferrous, under $1\frac{1}{4}$ in.	0.05 in.
Plain end ferrous, $1\frac{1}{4}$ in. and over.	0.065 in.
Plain end nonferrous.	0.0 in.

TABLE 12.3

ALLOWABLE *S* VALUES OF STRESS FOR REFRIGERATION PIPE⁵

Seamless Steel Grade A.	12,000 psi
Seamless Steel Grade B.	15,000
Lap Welded Steel.	9,000
Butt Welded Steel.	6,800
Electric Resistance Welded Steel.	10,200
Lap Welded Wrought Iron.	8,000
Butt Welded Wrought Iron.	6,000
Seamless Red Brass.	4,700
Copper.	4,000

Ordinary ferrous metals are not to be used below -60°F , and other materials must meet special tests. Steels of 3 to 4 per cent nickel are generally acceptable for quite low temperatures, and 18 per cent chrome, 8 per cent nickel steels are satisfactory for very low temperatures.⁶

In order to comply with ASA Codes B31-1 and B9, the minimum design pressure should be two-thirds of the test pressure given in Table 12.4. These codes specify:

(a) Every refrigerant containing part of every system shall be tested and proved tight by the manufacturer at not less than test pressures in Table 12.4.

(b) Every refrigerant-containing part of every system that is erected on the premises, except compressors, safety devices, pressure gauges and control mechanisms, that are factory tested, shall be tested and proved tight after complete installation and before operation at not less than test pressures in Table 12.4.

(c) No oxygen or any combustible gas or mixture shall be used for testing. [Carbon dioxide or other dry inert gas should be used.]

12.3. Joints and Fittings. Flanged, screwed, soldered (sweated capillary) or flared compression-type joints may be used with nonferrous piping. Flanged, screwed, or welded-type joints are used for ferrous pipe.

Welding has been gaining universal acceptance and is displacing the use of flanged and screwed fittings. Butt-welding fittings are used for nominal sizes of 2 in. and larger; socket-welding fittings are used for sizes under 2 in. Rules and procedures are given in the Piping Code.

⁵ From American Standard Code for Pressure Piping, ASA B31.1. New York: American Society of Mechanical Engineers, 1942, Table 42.

⁶ Joint ASME-ASTM Research Committee on the Effect of Temperatures on the Properties of Metals, *Impact Resistance and Tensile Properties of Metals at Sub-atmospheric Temperatures*. American Society for Testing Materials, August, 1941.

TABLE 12.4
TEST PRESSURES⁷

Refrigerant	Chemical Formula	Minimum Test Pressure, psi	
		High-Pressure Side	Low-Pressure Side
Ammonia.....	NH ₃	300	150
Butane.....	C ₄ H ₁₀	90	50
Carbon dioxide.....	CO ₂	1500	1000
Freon-12.....	CCl ₂ F ₂	235	145
Freon-114.....	C ₂ Cl ₂ F ₄	80	50
Carrene No. 1.....	CH ₂ Cl ₂	30	30
Freon-21.....	CHCl ₂ F	70	50
Dichloroethylene.....	C ₂ H ₂ Cl ₂	30	30
Ethane.....	C ₂ H ₆	1100	600
Ethyl chloride.....	C ₂ H ₅ Cl	60	50
Isobutane.....	(CH ₃) ₃ CH	130	75
Methyl chloride.....	CH ₃ Cl	215	125
Methyl formate.....	HCOOCH ₃	50	50
Propane.....	C ₃ H ₈	325	210
Sulfur dioxide.....	SO ₂	170	95
Freon-11.....	CCl ₃ F	50	30
Others: Use sat. pressure at.....		150 F	115 F

Screwed joints, including screwed-on flanges, are limited to not over 3 in. nominal size for pressures up to 250 psi, not over 1½ in. for pressures above 250 psi, and not over 4 in. for brine. Extra-heavy fittings are required for copper and brass. Exposed threads should be tinned or otherwise coated to prevent corrosion.

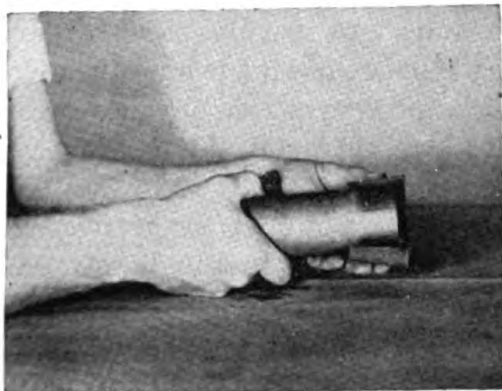
Flanged joints have been used for pipe of all sizes. Standards have been proposed but have not been agreed upon by the manufacturers, so that variations in dimensions are found. The most recent development of a standard for ammonia flanged fittings has been dropped because of the decreased use of ammonia and the increased use of welding for steel pipe. The Piping Code specifies that for refrigeration, flange faces shall be of the retained gasket type, and manufacturers' standards are of the tongue-and-groove type.

Flared compression fittings are allowed only for annealed or soft copper tubing not over ¾ in. O.D., with the additional condition that all joints be visible for inspection. This type of joint is easily and quickly made and is used extensively in domestic and commercial refrigeration

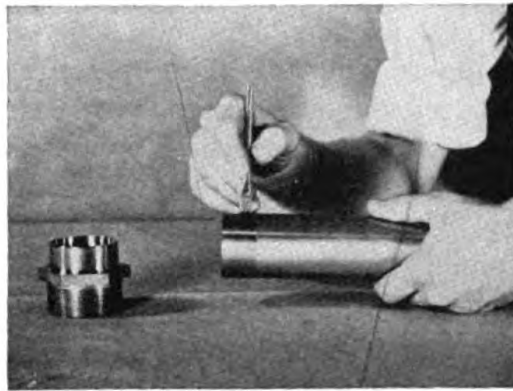
⁷ From American Standard Code for Pressure Piping, ASA B31.1. New York: American Society of Mechanical Engineers, 1942, Table 41.

installations. The fittings are of forged brass and are made in accordance with SAE standards.

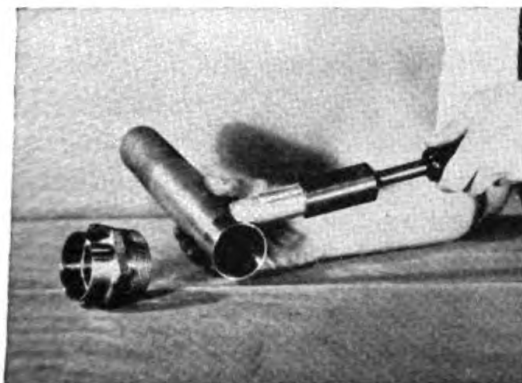
When any copper tubing over $\frac{3}{4}$ in. is used on an installation, all the joints are usually soldered. Hard-temper tubing is used to make a neat appearance. Fittings of cast red brass or die-pressed, wrought, or extended brass or copper may be used, but wrought copper is preferred.



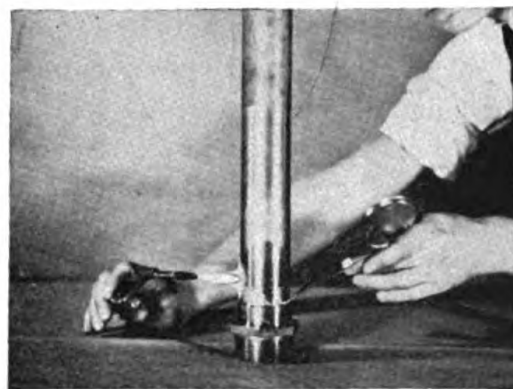
(a)



(b)



(c)



(d)

Fig. 12.1. Four Steps in Making a Solder Joint: (a) Cleaning both tube and fitting. (b) Applying flux to tube and fitting. (c) Tinning tube and fitting. (d) Heating and adding solder to fill joint. Courtesy Crane Co.

The joints are a socket-sleeve or sweated capillary type with a clearance, caused by a diameter difference, of from 0.002 to 0.009 in. It is very important that both surfaces be clean and polished and that a good flux be used. The joints are then put in place, well heated with a torch, and the solder applied. The solder flows by capillary action over the surfaces, bonding them together. Figure 12.1 shows the four steps recommended for soldering stainless steel tubes.⁸ These steps apply equally to soldering copper tubing. Soft solders, 50-50 tin and lead or the better 95-5 tin and

⁸ Seabloom, Eric R., "Welding Alloy Steel Piping." *Heating, Piping and Air Conditioning*, Vol. 18, No. 12 (December, 1946), p. 79.

lead or tin and antimony with melting points under 450 F, and hard solders, silver alloys with melting points over 1000 F, are used. Most shop joints are silver-soldered or brazed. The Piping Code specifies hard or silver solder on tubing over $\frac{5}{8}$ in. O.D. or if over 20 lb of refrigerant is in the system.

12.4. Supports and Expansion. The pipe connecting the units must be supported securely to minimize vibrations and stresses on the joints. Metal hangers, brackets, straps, or clamps are usually used. These supports should be at intervals not over 40 times the outside pipe diameter and should not restrain thermal expansion and contraction or cause excessive stresses at critical points. Codes require metal sleeves on pipe passing through any building structure material.

Although slip-type expansion joints are prohibited for refrigerant piping, a refrigerating system must still be designed so that undue stresses from temperature change are not created. Flexibility is usually obtained by bends or offsets or by installing at least three elbows, even in the shortest lines, when hard-temper tubing or rigid pipe is used. Copper and brass expand about 50 per cent more than steel or iron, but the temperature differences encountered in refrigeration are usually not so great as those in steam piping.

12.5. Friction. The pressure drop or friction in straight pipe can be calculated as explained in Chapter 8. Fittings also cause pressure drop. For handling water or similar liquids, the resistance of each fitting is expressed in an equivalent number of elbows having the same pressure drop. The resistance of an elbow is approximately equal to that for a pipe of the same size having a length 25 times the diameter. More accurate data is given in the *Piping Handbook*.⁹ In handling refrigerants, the resistance of all fittings is usually expressed as that of an equivalent length of the same size pipe. Values for fittings of standard-weight pipe and of Type L copper tubing are given in Appendix Table A.22. The actual pipe length is added to the total equivalent length of all the fittings in the line to obtain the total equivalent length, which is then used to determine the total pressure drop for the line.

Elbows with a radius of curvature from 2 to 4 times the pipe diameter have a minimum pressure drop. Radii of 4 to 6 diameters are very little better than radii of 1.5 to 2 diameters. In the longer radius elbows the disturbance is less but occurs through a longer length.

12.6. Sudden Velocity Changes. A sudden change in the cross-sectional area across which a fluid is flowing causes a change in velocity accompanied by a loss in available pressure. The energy losses when the velocity decreases are generally greater than when it increases. The loss due to a sudden expansion is

⁹ Crocker, S., *op. cit.*, pp. 95-102.

$$\Delta p = \frac{(V_1 - V_2)^2 \rho}{2g} = \left(1 - \frac{d_1^2}{d_2^2}\right)^2 \frac{V_1^2 \rho}{2g} \quad \text{psf} \quad (12.2)$$

where subscripts 1 and 2 refer to the small and large cross-sectional areas, respectively.

The loss due to a sudden contraction is about 0.7 of the loss for a sudden expansion. It can be calculated from the proposed formula¹⁰

$$\Delta p = \left(\frac{1}{C} - 1\right) \frac{V_2^2 \rho}{2g} \quad \text{psf} \quad (12.3)$$

where

$$C = 0.582 + \frac{0.0148}{1.1 - d_2/d_1}$$

12.7. Liquid Lines. The refrigerant flowing from the condenser to the expansion valve is normally all in the liquid state, and it is desirable

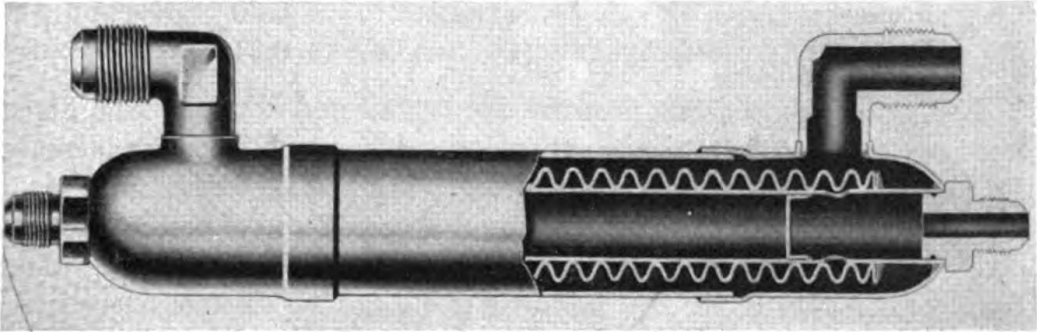


Fig. 12.2. Liquid-suction heat exchanger. Courtesy Superior Valve & Fittings Co.

that it be slightly subcooled. When a separate receiver and condenser are used, two lines are involved. In such cases the receiver is usually close to the condenser, and the short interconnecting line is installed at the factory on the modern equipment. Even on systems whose condenser and receiver are assembled in the field, the connecting pipe is not very long. In any event, it should be of ample size and sloped down toward the receiver from the bottom of the condenser to prevent its becoming gas-bound. A velocity under 120 fpm is recommended.

The line from the receiver to the expansion valve is best designed for a pressure drop of under 5 psi, with 10 psi as the maximum to be considered. A velocity between 100 and 250 fpm is recommended. The pressure drop should not be so great that the reduction in refrigerant pressure can cause vaporization ahead of the expansion valve. Of course a loss in static head or pressure, due to the evaporator being at a higher elevation than the condenser and receiver, also can cause vaporization.

When there is a possibility that vaporization of some of the liquid will occur ahead of the expansion valve, means for cooling the liquid should

¹⁰ Crocker, S., *op. cit.*, p. 103.

be provided. On some installations the suction line from the evaporator is made larger than for normal design and the liquid line is run inside it. On other installations a special liquid-suction heat exchanger such as shown in Figs. 12.2 and 11.18 is used. These devices help keep the liquid-line refrigerant subcooled and the suction-line vapor superheated, both desirable procedures. The magnitude of the drop in liquid-line pressure for each foot of static liquid head is not always appreciated. Approximate values are as follows:

Ammonia.....	0.26 psi per ft
Carbon dioxide.....	0.33
Sulfur dioxide.....	0.60
Methyl chloride.....	0.40
Freon-11.....	0.64
Freon-12.....	0.57
Freon-21.....	0.60
Freon-22.....	0.51

Keeping the liquid-line pressure drop small makes available a higher pressure at the entrance to the expansion valve, which in return increases the capacity of the valve.

Test results on pressure drop of liquid Freon-12 and of methyl chloride have been reported and compared favorably with calculated values, assuming 0, 5, and 10 per cent mixtures of oils having various viscosities.¹¹ The calculations indicate, for example, that in a $\frac{5}{8}$ -in.-O.D., 90 F Freon-12 line for 10 tons, 5 per cent of 500 Saybolt oil increases the pressure drop from 7 psi for pure Freon-12 to 16 psi per 100 ft. With 10 per cent oil the pressure drop is 24 psi per 100 ft. For the same conditions the effect of temperature with 5 per cent oil is indicated by the following values of pressure drop: 15 psi at 80 F, 16 psi at 90 F, and 18 psi at 100 F, all per 100 ft of pipe.

If the receiver is above the evaporator, a vertical loop should be installed 6 ft up, directly after leaving the receiver, to prevent liquid from siphoning into the evaporator through leaking valves during shutdown. Liquid-line solenoid valves are also recommended.

12.8. Suction Lines. More careful consideration must be taken in designing and sizing the vapor refrigerant line from the evaporator to the compressor than for the other lines. As previously pointed out, a reduction in suction pressure at the compressor, with other factors held constant, means an appreciable reduction in capacity and more power input per ton of refrigeration. Therefore, the pressure drop in the suction line should be a minimum so that the suction pressure at the compressor will be as near the required suction pressure at the evaporator as is practical.

¹¹ Gygax, E. and Willson, K. S., "Pressure Drop in Refrigerant Liquid Lines." *Refrigeration Engineering*, Vol. 39, No. 2 (February, 1940), pp. 103-106.

Comparatively high velocities, 500 to 5000 fpm, may be used. The optimum value depends upon the refrigerant and the operating pressure range and usually is between 1000 to 2000 fpm for the Freons. When the Freons or methyl chloride are used, the return of oil to the compressor must be considered. For these refrigerants, even at light loads, the minimum velocity shall be 500 fpm for horizontal runs and 1000 fpm for vertical runs. Too high a velocity, however, creates noise problems so that the maximum desirable velocity for a heavy vapor like carbon dioxide is about 1300 fpm. It is recommended that the suction line on the ordinary installation be designed for a total pressure drop of about 1 to 2 psi if the velocity can be kept within the limits specified.

Since the properties of a refrigerant vapor change appreciably with a change in pressure and temperature, the suction line must be designed for the proper operating conditions. Table 12.5 shows the approximate

TABLE 12.5
APPROXIMATE SUCTION-LINE CAPACITY FACTORS FOR EQUAL PRESSURE DROP OF
FREON-12¹²

Saturated suction temperature	50	40	30	20	10	0	-10	-20
Factor	1.09	1.00	0.92	0.86	0.80	0.74	0.66	0.56

comparative capacities that can be carried with the same pressure drop in a given suction pipe at various saturation temperatures.

Figure 12.3 shows recommended methods of installing the liquid and suction lines when the evaporator is (a) above the compressor, (b) on the same level, and (c) below the compressor. Figure 12.4 shows suction piping for multiple evaporators (a) above one another and connected to a compressor above them, (b) above one another and connected to a compressor below them, and (c) all on the same level. Trapping the evaporator helps prevent liquid slugs from reaching the compressor. In multiple evaporators it also prevents possible liquid and oil slopover of one evaporator from affecting the operation of the thermostatic expansion valve's thermal bulb on another evaporator. The thermal bulb should be installed near the top of a horizontal section of the suction line on the coil side of any possible oil trap.

Results of tests to determine the pressure drop in Freon-12 and methyl chloride suction lines,¹³ were found to check against calculations using

¹² Determined from values in Table 5, *ACRMA Equipment Standard 520*. Washington, D.C.: Air Conditioning and Refrigerating Machinery Association, 1946.

¹³ Gyax, E., Willson, K. S., and Clark, G. H., "Pressure Drop in Suction Lines." *Refrigeration Service Engineer*, December, 1938.

equation 8.3 with values of the friction factor f taken from a curve plotted in terms of a slightly modified Reynolds number.

12.9. Discharge Lines. Discharge lines, sometimes called *hot gas lines*, refer to refrigerant connections between the compressor and the

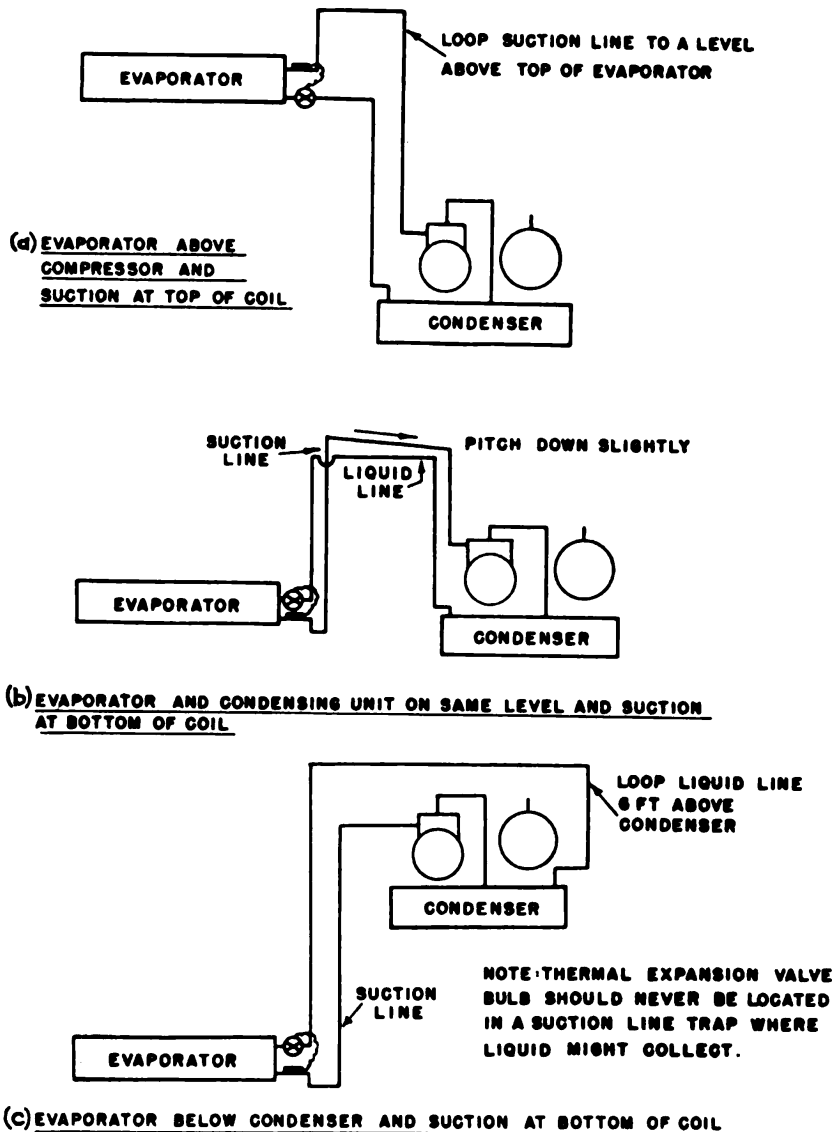


Fig. 12.3. Suction and liquid-line piping.

condenser. Air-cooled, most water-cooled, and some evaporative-cooled condensing units are factory-assembled as a complete self-contained unit including motor, compressor, and condenser. The discharge line is then a part of the assembly, designed and installed by the manufacturer. When a separate or remote condenser is used, the discharge line is installed on the job.

As in suction-line design, an increase in pressure drop in the discharge line means that the compressor must operate between a greater pressure

differential and will require more horsepower per ton of refrigeration. However, the effect per unit of pressure increase at the discharge of a compressor is not so great as the same unit of pressure decrease at the suction of the compressor. The actual comparison may be illustrated

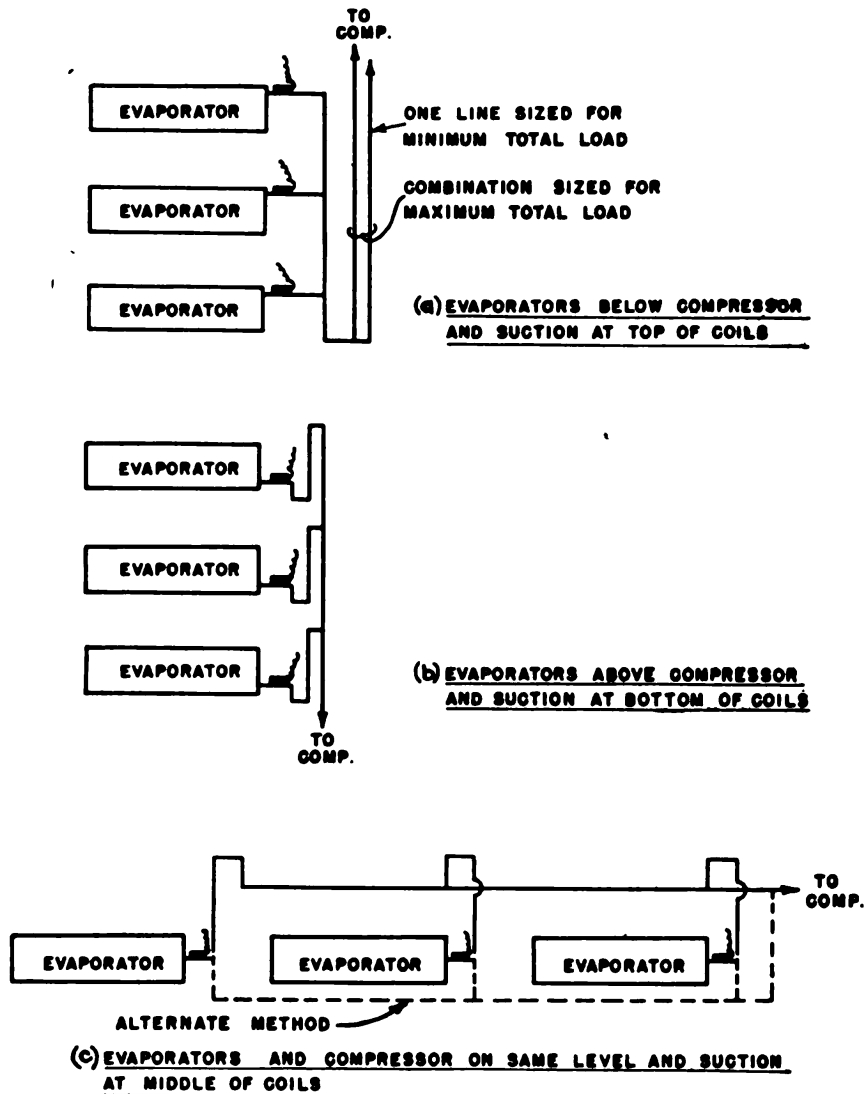


Fig. 12.4. Multiple-evaporator suction-line piping.

by plotting the published ratings of a Freon-12 refrigeration compressor as shown in Fig. 12.5. The ratings for a machine operating at a constant speed indicate that the capacity at constant condensing or discharge varies much more per psi change in suction pressure than the capacity at constant suction pressure varies per unit change in discharge pressure. The difference in the horsepower per ton variation with psi pressure change, though not so great, still indicates that a change in suction pressure has more effect than a change in discharge pressure for the range

shown. This fact further emphasizes the importance of proper suction-line sizing as stressed in the preceding section.

The velocity range in discharge lines is 1000 to 5000 fpm, except for carbon dioxide, where noise limits the maximum to about 1300 fpm. A pressure drop of 2 to 4 psi is recommended for the discharge line.

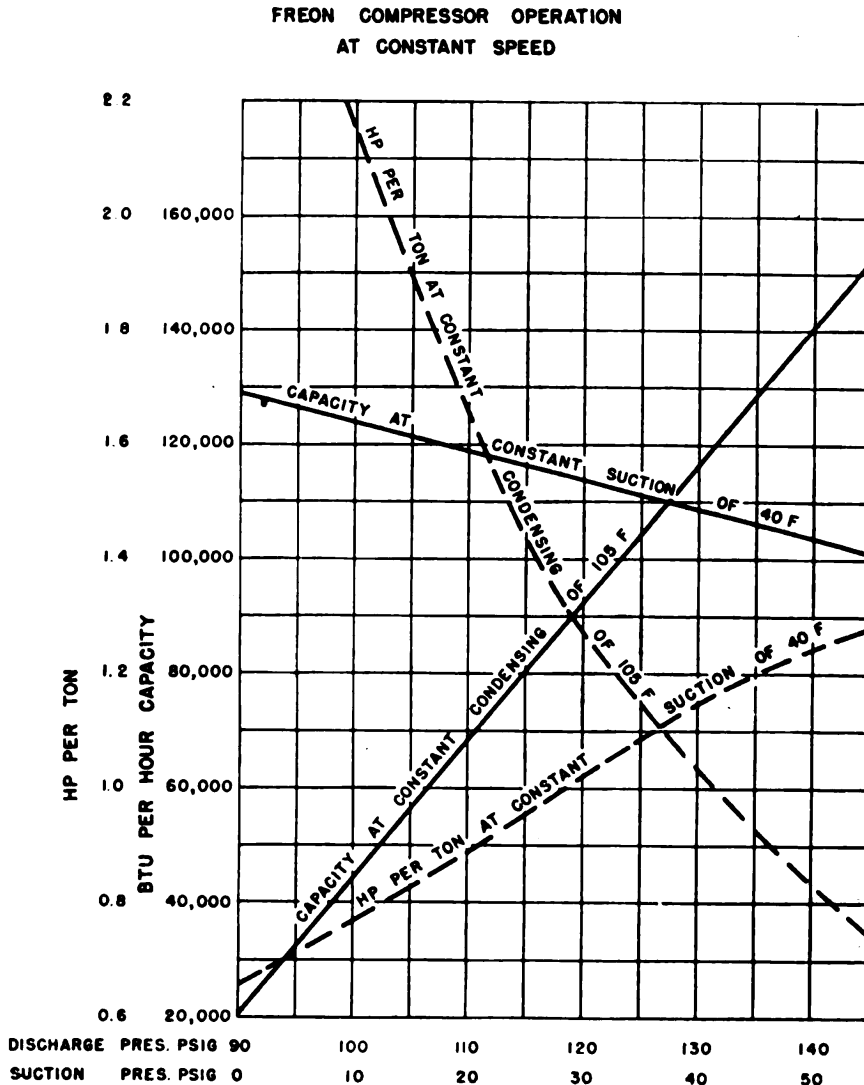


Fig. 12.5. Effect of compressor discharge and suction pressure variations on capacity.

If the condenser must be above the compressor or in a space at a temperature higher than that occupied by the compressor, a vertical "U" loop should be installed directly after the pipe leaves the compressor and before it rises. If the condenser inlet is at the bottom, a loop must be made up to the top of the condenser before the line enters it. These precautions prevent liquid from draining into the compressor during shutdown.

12.10. Pipe-Size Selection. Refrigerant lines of proper size should

be used instead of pipe or tubing arbitrarily selected to match the connecting fittings furnished with the various pieces of equipment. Appendix Tables A.23 through A.26 give pressure-drop data for the commonly used refrigerants. In general, line sizes for Freon-22 are smaller than for Freon-12 and larger than for methyl chloride. Ammonia lines are smaller than those for other commonly used refrigerants.

EXAMPLE 12.1. Determine (a) the sizes of Type L copper tubing to use for a 10-ton Freon-12 system operating at 110 F condensing and 40 F suction temperatures for the liquid, suction, and discharge lines and having the lengths and fittings indicated, and (b) the same but at 90 F condensing and 0 F suction.

<i>Discharge Line</i>	<i>Liquid Line</i>	<i>Suction Line</i>
20 ft	20 ft	20 ft
4 elbows	4 elbows	4 elbows

SOLUTION:

(a) From Appendix Tables A.25 and A.26, $1\frac{5}{8}$ -in.-O.D. discharge, $\frac{7}{8}$ -in.-O.D. liquid, and $1\frac{5}{8}$ -in.-O.D. suction appear satisfactory. By Table A.22 the equivalent length of elbows would then be 16 ft for the discharge, 8 ft for the liquid, and 16 ft for the suction line.

With 10 degrees subcooling and with saturated vapor leaving the evaporator, the rate of refrigerant flow is

$$10 \times \frac{200}{82.71 - 31.16} = \frac{2000}{51.55} = 38.8 \text{ lb per min}$$

At 40 F evaporator the pressure drop for the suction line is

$$4.1 \text{ psi} \times \frac{36'}{100'} = 1.48 \text{ psi for 36 ft}$$

The pressure drop increases about as the square of the rate of flow. Converting from the 10.5 ton capacity given in Table A.25 for 3 psi drop, the pressure drop in the liquid line will be

$$\left(\frac{10.0}{10.5}\right)^2 \times 3 \text{ psi} \times \frac{28}{100} = 0.76 \text{ psi for 28 ft}$$

(b) At 90 F condensing and 0 F suction the discharge line should be $2\frac{1}{8}$ in. O.D. and the liquid line $\frac{7}{8}$ in. O.D. By Table 12.5, the tonnage load to use for suction-line sizing would be $10 \div 0.74 = 13.5$ tons. According to the suction-line table, A.26, a $2\frac{1}{8}$ -in.-O.D. line is required. Thus, larger suction and discharge lines should be used for these conditions than for those in (a).

12.11. Moisture Removal. Although most refrigerating equipment and tubing is dehydrated at the factory and shipped with the connections plugged, capped, or sealed, some air and moisture enters the system during installation. It is very important that all the moisture be removed from the system no matter what the operating temperatures are to be. This

precaution is particularly to be observed if the evaporator is to operate at a temperature below 32 F, where freezing at the expansion valve would restrict or stop the refrigerant flow. The majority of compressor seal failures have been attributed to the presence of moisture. Air and moisture in a Freon or methyl chloride system cause a hydrolysis, resulting in corrosion of steel parts and copper plating of seals and pistons.

After the equipment is installed, the piping completed, and the system purged and tested for leaks, it is recommended that a good high-vacuum pump be connected into the system and the entire refrigerant circuit evacuated. Care should be taken to note that all valves in the circuit are open. Not only air is drawn out by this process; any liquid water present vaporizes and is drawn out if sufficient time is allowed. When a high vacuum, 0.2 in. Hg or lower absolute pressure, has been attained, the pump may be shut off and the connection to it closed. The low pressure should be retained in the system. If it is not, some liquid is still vaporizing or there are leaks in the system. This procedure should be repeated or corrections made until a vacuum can be held. Additional details of the procedure may be found in service manuals.

Many contractors have relied upon the insertion of chemical driers, installed after purging the system, to remove the remaining moisture and have not used vacuum pumps. This procedure is not recommended for systems of over 3 hp. When the evaporator temperature is to be below 32 F, a drier is sometimes installed even after a vacuum pump is used.

Driers are usually copper cylinders fitted with tubing connections at each end. The recommended desiccant charge is silica gel or activated alumina. The drier must be kept airtight until installed; otherwise the chemical will absorb moisture from the atmosphere and will soon become saturated and ineffective. Typical dehydrators are shown in Fig. 12.6. Screens surround the tubing connections. The desiccant can be replaced through the plugged top opening in the drier in Fig. 12.6a and through the bolted cap on the end of the drier in Figure 12.6b.

12.12. Water Piping. Water is often needed in part of a refrigerating system. In most installations it serves as the cooling medium for the condenser. In some it is also chilled and pumped to the cooling coils. When used for condensing, water is piped from city mains, from wells, from a reservoir, to and from a cooling tower, or through an evaporative-condenser spray circuit. When the system is closed and relatively simple, which is usually the case, a water velocity from 3 to 10 fps proves most economical. When the allowable pressure drop is limited, the piping must be designed accordingly.

Iron pipe is usually used, and the water friction chart in the Appendix, Fig. A.4, may be applied. For pump systems it is recommended that the pipe be sized for a pressure drop under 2 psi per 100 ft. and at a velocity under 6 fps. When water is taken from city mains, the piping should

be designed so that the pressure drop is not great enough to cause too small a flow during periods of minimum available pressure.

12.13. Brine Piping. When a brine is chilled and serves as the cooling medium, it must be pumped through a pipe system. Since the density

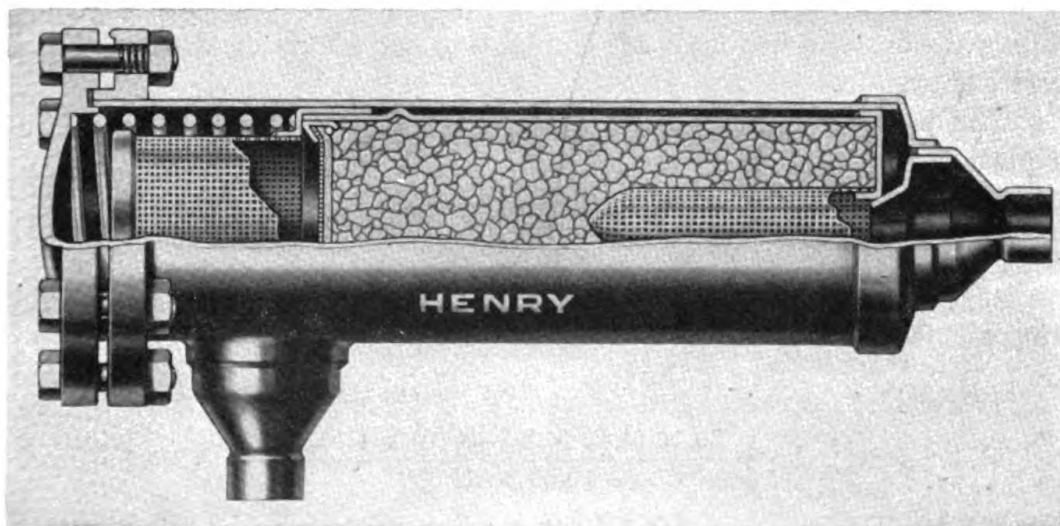
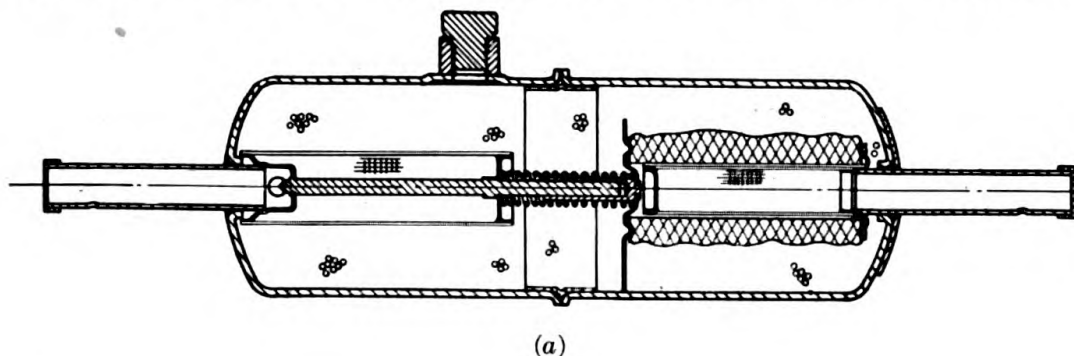


Fig. 12.6. Refrigerant-line dehydrators. (a) Courtesy Automatic Products Co.
(b) Courtesy Henry Valve Co.

of brine is greater than that of water, the frictional resistance is greater. The usual procedure in sizing brine pipe is to use brine friction multipliers with water friction charts or tables. Figures A.5 and A.6 in the Appendix give suggested brine multipliers to be used. The most economical brine velocity is between 4 and 7 fps.

EXAMPLE 12.2. What size pipe should be recommended for handling 30 gpm of (a) water and (b) 10 F calcium chloride brine with a specific gravity of 1.20? Each will flow in a 50-ft line having 1 globe valve and 6 elbows. What would be the approximate pressure drop?

SOLUTION:

(a) From Fig. A.4 in the Appendix a $1\frac{1}{2}$ in. pipe indicates a pressure drop of 3.8 psi per 100 ft. From Appendix Table A.22 a globe valve equals 42 equiva-

lent feet and an elbow 4 equivalent feet. Therefore the equivalent length is

$$50 + 42 + 6 \times 4 = 116 \text{ ft}$$

For water the pressure drop would be $3.8 \times \frac{1.16}{100} = 4.41 \text{ psi}$

(b) For the brine, from Fig. A.6, the multiplier is 1.44. The pressure drop then would be

$$1.44 \times 3.8 \times \frac{1.16}{100} = 6.35 \text{ psi}$$

12.14. Pumps. Since most water or brine systems in refrigeration work are not too complicated, the proper selection of a pump is not difficult. When a pump is ordered, the gpm and head must be specified. The gpm is determined when selecting or designing the equipment in the system. The head is determined by first computing the total equivalent length of pipe and fittings and multiplying by the friction loss per unit length. To this, for an open system, must be added the vertical lift. For a closed system, provision or allowance must be made for originally filling the system, including air elimination and elevating the liquid in a vertical system.

The power required to drive a pump may be computed from the equation

$$\text{hp} = \frac{\text{gpm} \times \text{ft head} \times \text{sp gr} \times 8.33}{\eta \times 33,000} \quad (12.4)$$

EXAMPLE 12.3. What pump power would be required in Example 12.2 if the pump efficiency is 60 per cent and there is a vertical lift of 10 ft?

SOLUTION:

$$(a) \text{ for water: } \text{hp} = \frac{30 \times (4.41 \times 2.3 + 10) \times 1 \times 8.33}{0.60 \times 33,000} = 0.254 \text{ hp}$$

$$(b) \text{ for the brine: } \text{hp} = \frac{30 \times (6.35 \times 2.3 + 10) \times 1.20 \times 8.33}{0.60 \times 33,000} = 0.373 \text{ hp}$$

12.15. Accessories. Valves of various types and strainers and sight glasses are installed in refrigerant lines. Expansion valves, solenoid valves, and pressure regulating valves are explained in Chapter 14. All these accessories usually are made with sweat-type solder connections for use in copper-tubing lines or with threaded connections for use in steel or other threaded pipe lines. If these solder connections are lacking, adapters (screwed to soldered fittings) must be used. During installation of thermal expansion valves or other fluid-operated valves, the assembly containing the fluid should be removed to prevent possible damage from overheating with the torch.

The use of hand shutoff valves should be kept at a minimum since they increase the possible sources of leaks. They are used with multiple units or dual piping where it is necessary to close off part of the system occasionally. Both packed and packless valves are available. Many

engineers prefer the packless type because of reduced leakage around the valve stem. Typical valves are shown in Figs. 12.7 to 12.13.

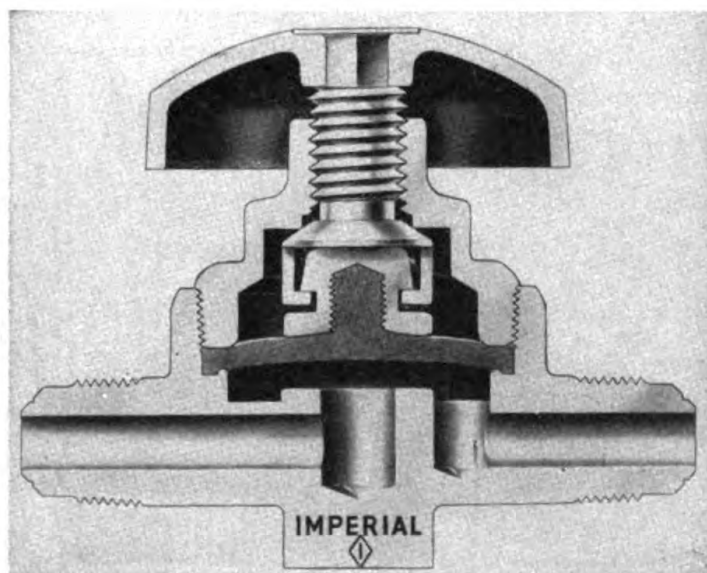


Fig. 12.7. Packless-diaphragm refrigerant-line valve. Courtesy Imperial Brass Mfg. Co.

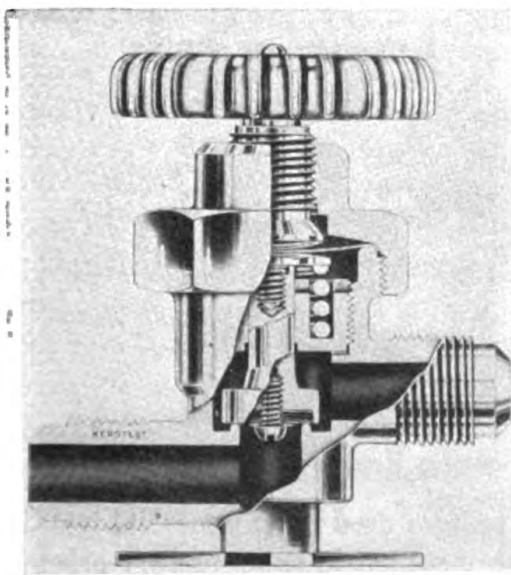


Fig. 12.8. Offset packless valve. Courtesy Kerotest Mfg. Co.

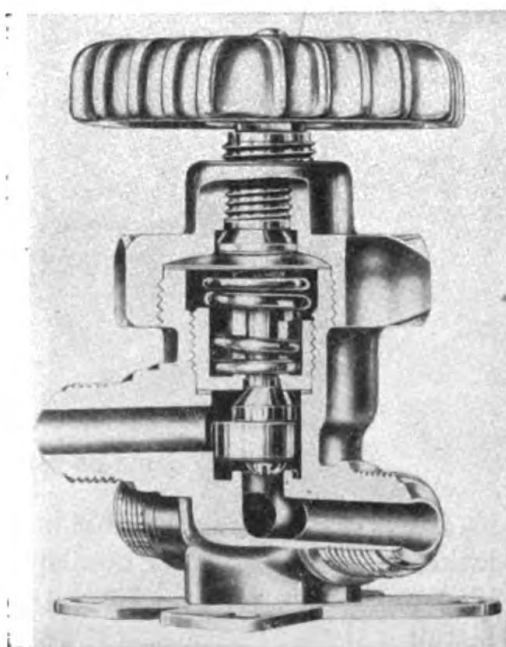


Fig. 12.9. Three-way packless valve. Courtesy Kerotest Mfg. Co.

Strainers, preferably of the Y type, in which the screen can be removed without dismantling the piping, should be installed just ahead of the expansion valves or solenoid valves in the liquid-refrigerant line. One

strainer per evaporator is usually sufficient. A strainer should also be placed in the water line just ahead of the water regulating valve when a water-cooled condenser is used. A typical strainer is shown in Fig. 12.14.

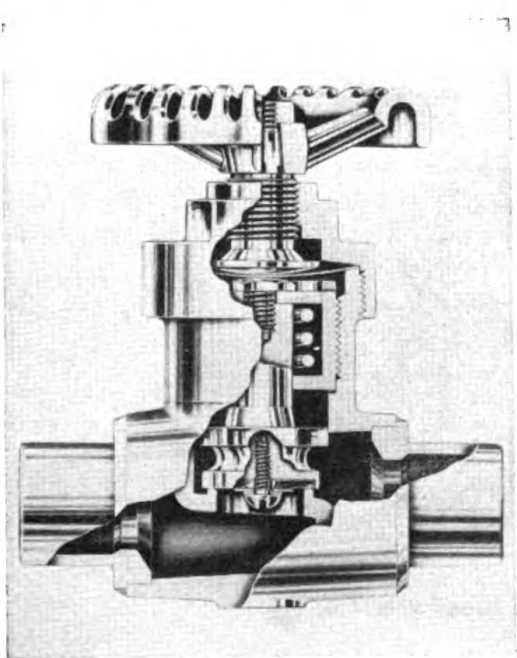


Fig. 12.10. Packless globe valve. Courtesy Kerotest Mfg. Co.

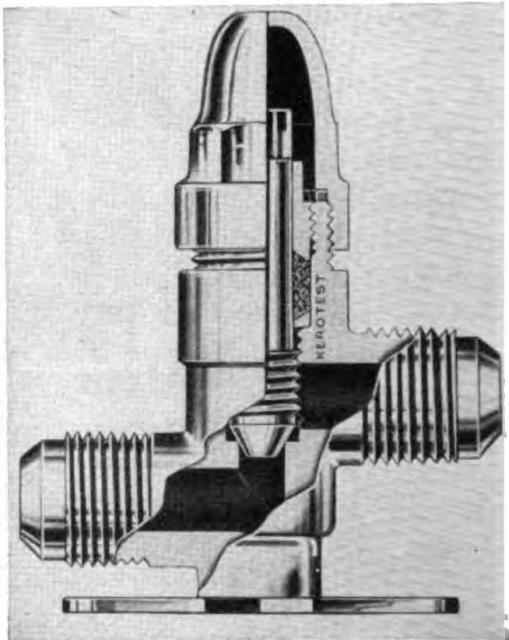


Fig. 12.11. Packed offset valve. Courtesy Kerotest Mfg. Co.

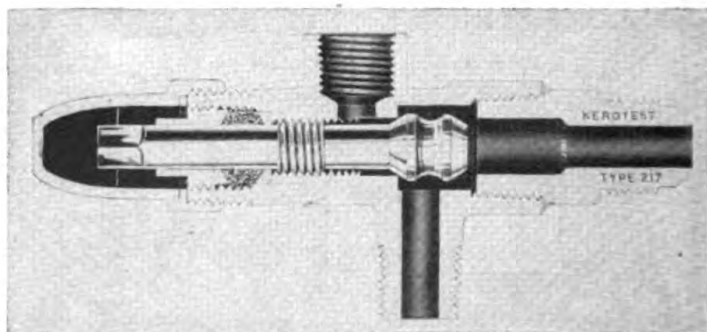


Fig. 12.12. Shut-off valve. Courtesy Kerotest Mfg. Co.

Sight glasses are available but are seldom used on commercial installations. However, they now are being installed in some factory-assembled units, as shown in Figs. 18.1 and 18.3. It is always advisable to install a sight glass in the liquid line just ahead of the expansion valve when the evaporator is above the condenser. It then is easy to see whether any of the liquid has vaporized before entering the expansion valve.

Vibration eliminators are available for connection in copper lines where there is a possibility of pipe vibration, as when the compressor is mounted on a floating or springy base. Vibration eliminators usually

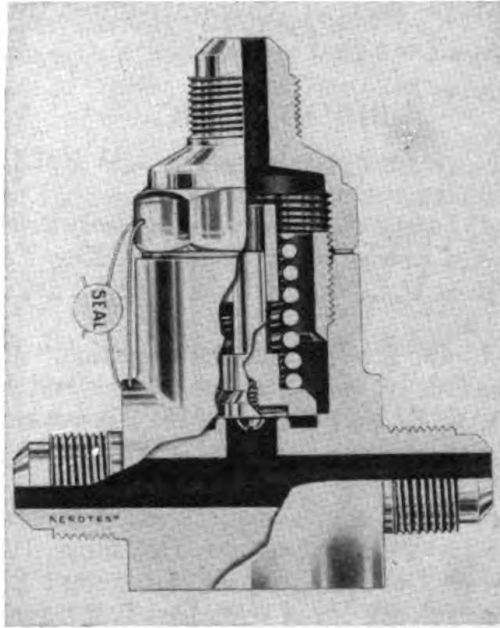


Fig. 12.13. High-pressure relief valve.
Courtesy Kerotest Mfg. Co.

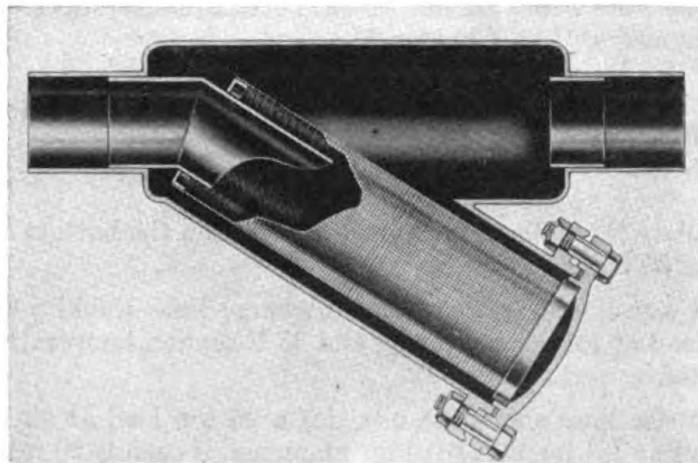


Fig. 12.14. Refrigerant liquid-line strainer. Courtesy
Henry Valve Co.

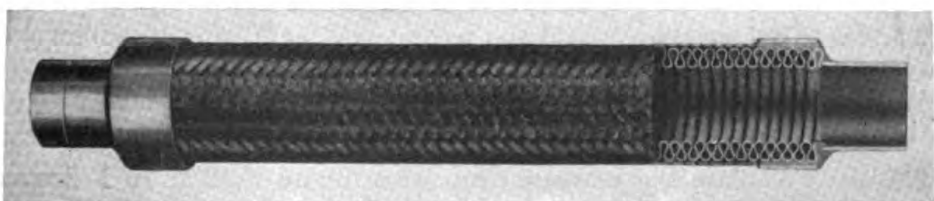


Fig. 12.15. Vibration eliminator. Courtesy American Metal Hose Branch
of American Brass Co.

consist of a bellows-type section of copper-alloy tubing covered with woven copper wire, as shown in Fig. 12.15.

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PROBLEMS

12.1. Calculate the difference in pressure between the bottom and top of an F-12 liquid line 20 ft high at 70 F.

12.2. What size liquid, suction, and discharge lines would you recommend for a 100 ton load at 110 F condensing and 33 F suction temperatures if each is 50 ft long? Refrigerant is Freon-12.

12.3. What size lines should be used for a 20 ton load at 90 F condensing and -20 F suction (a) for F.12? (b) for ammonia, if each is 20 ft long?

12.4. A $\frac{7}{8}$ -in.-O.D. liquid line carries F-12, which is subcooled 10 F and is at 70 F, for 30 ft with 4 elbows and 1 globe-type valve in the line. Define the state of the refrigerant by stating how much subcooling or what per cent vapor would be in the line at the end of 30 ft when the end is 15 ft above the starting point. Capacity of system is 10.5 tons.

12.5. (a) What size pipe would you recommend for 20 gpm of sodium chloride brine (specific gravity 1.10) flowing for 100 ft with 8 elbows and one angle valve? (b) What would the pressure drop be? (c) What hp motor would be required if both the pump and motor are 50 per cent efficient? Brine is at 20 F.

12.6. (a) Calculate the pressure drop when liquid F-12 at 70 F flows from a $\frac{7}{8}$ -in.-O.D. pipe into a $2\frac{1}{8}$ -in.-O.D. pipe, if the velocity in the smaller pipe is 1.0 fps. (b) Calculate the pressure loss on reentering a $\frac{7}{8}$ -in. pipe from the $2\frac{1}{8}$ -in. tube.

12.7. Determine the discharge-, liquid-, and suction-line sizes for a 15-ton methyl chloride system with a 20 F evaporator and 86 F condenser, if the equivalent length of each line is 30 ft. Could F-12 at the same conditions be used in these lines? Compare values.

12.8. What size chilled-water pipe would you recommend to carry 60 gpm in a closed system, if there are 80 ft of pipe, 8 elbows, 2 globe valves, a coil, a chiller, and a pump? The resistance of the coil and chiller totals 20 ft of water.

12.9. What size water condenser pipe should be used if the minimum city pressure is 30 psi? The pipe is 90 ft long with 9 elbows and 1 globe valve. The resistance of the water regulator and condenser for the 15 gpm required is 20 psi.

12.10. How much water will flow through a 1-in. steel pipe 100 ft long with 10 elbows and 3 globe valves, if there is a 10-ft lift and 20 psi pressure is available at the start?

PART IV

Advanced Refrigeration Topics

CHAPTER 13

Multiple Evaporator and Compressor Systems

13.1. Applications. In many refrigeration installations different temperatures are required at various points in the plant. This is true in various industrial plants and in hotels, institutions, large restaurants, and food markets where food products of all kinds are received in large quantities and stored at different temperatures. For example, the different rooms in an establishment requiring the storage of fresh fruits, fresh vegetables, fresh-cut meats, frozen products, dairy products, canned goods, bottled goods, and ice cannot all be maintained at the same temperature and humidity. Water chillers, display cases, or other cooling equipment might also be included.

Generally each location is cooled by its own evaporator in order to obtain more satisfactory control and for economy of initial and operating costs. However, the answers are not so evident to the questions of whether one compressor should serve all evaporators, whether each evaporator should have a separate compressor, or whether an intermediate number of compressors should be used. Initial costs, operating costs, and space requirements must be considered. Comparative initial cost estimates can be obtained from equipment and installation estimates for the system under consideration. The theoretical power requirements necessary for an analysis of probable operating costs will be determined for several different combinations in the following sections.

Each classification of system will be analyzed for the same temperature and cooling-load requirements of a given plant. There would be variations in the comparative values for other temperatures or other loads, but the relative effectiveness of each system can be seen. It must be kept in mind that the following are theoretical analyses. Corrections for throttling at the valves, cylinder heat losses, the actual compression process, leakage, and other factors must be considered when dealing with actual equipment installations. In many cases, however, the theoretical comparison may be sufficient.

For comparative purposes assume that a plant is to use Freon-12 refrigerant for loads of 10 tons at 44 F evaporator temperature, 30 tons at 34 F, and 20 tons at 24 F; the condenser pressure is 139 psia; the liquid is subcooled 20 F; and the vapor leaves each evaporator dry-saturated. Isentropic compression will be assumed throughout. A pressure-enthalpy diagram for each system as well as a schematic piping diagram is included

with each analysis. Similar analyses for ammonia cycles are given in the *Refrigerating Data Book*.¹

13.2. One-Compressor Systems. Three methods of using one compressor for three evaporators will be considered.

All evaporators operating at the same temperature constitute one method of operation. However, this arrangement is not practical if food products

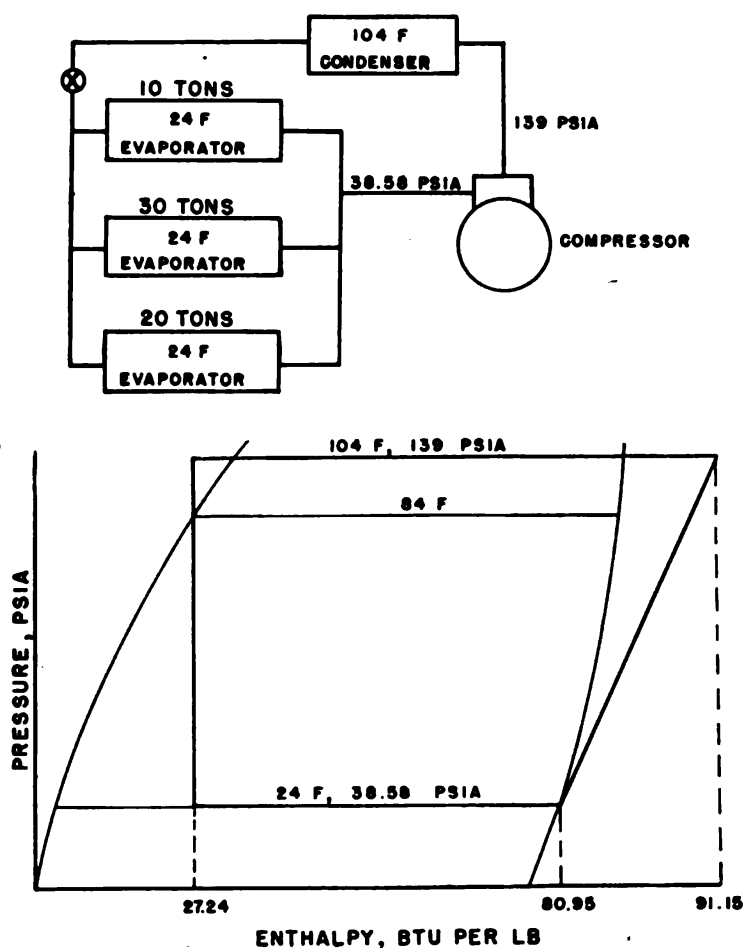


Fig. 13.1. All evaporators at same temperature.

or other hygroscopic materials are to be exposed in the higher temperature rooms, since the large temperature differential between the room air and coil may dehydrate the product. This system may be used where temperature control only is needed and where low humidity with no control can be tolerated. Such a system is diagrammed in Fig. 13.1.

The refrigerating effect in each evaporator is equal to the enthalpy difference (h_g at 24 F) — (h_f at 84 F) or $80.95 - 27.24 = 53.71$ Btu per pound. The quantity of refrigerant circulated is $(60 \times 200)/53.71 =$

¹ *Refrigerating Data Book*, 5th ed. New York: American Society of Refrigerating Engineers, 1943, Chapter 2.

223.4 lb per minute. The theoretical power required is $(91.15 - 80.95) 223.4/42.4 = 53.6$ hp.

Individual expansion valves and back-pressure valves are used in the second system, as shown in Fig. 13.2. The refrigerating effect in the 24 F evaporator is $80.95 - 27.24 = 53.71$ Btu per pound. The rate of refrigerant flow is $(20 \times 200)/53.71 = 74.5$ lb per minute.

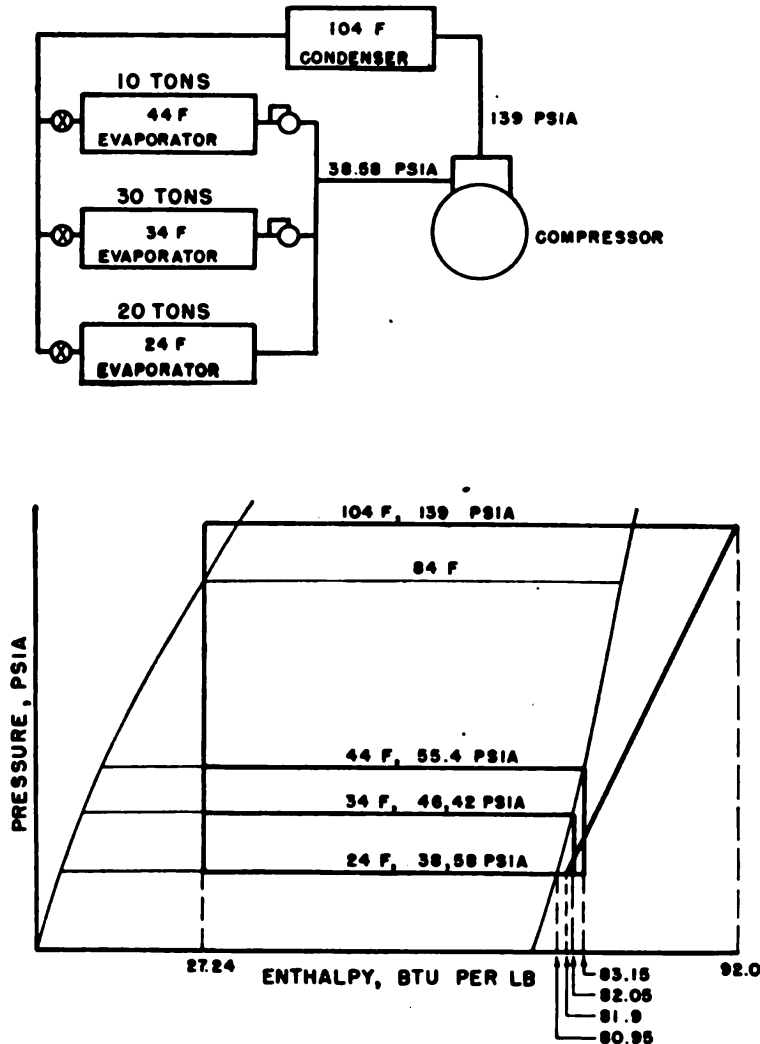


Fig. 13.2. Individual expansion valves and back-pressure valves.

In the 34 F evaporator the refrigerating effect is $(82.05 - 27.24) = 54.81$ Btu per pound. The rate of refrigerant flow is $(30 \times 200)/54.81 = 109.5$ lb per minute.

In the 44 F evaporator the refrigerating effect is $(83.15 - 27.24) = 55.91$ Btu per pound. The rate of refrigerant flow is $(10 \times 200)/55.91 = 35.8$ lb per minute. The total refrigerant flow is $74.5 + 109.5 + 35.8 = 219.8$ lb per minute. The enthalpy of the vapor mixture entering the compressor is $(74.5 \times 80.95 + 109.5 \times 82.05 + 35.8 \times 83.15)/219.8 =$

81.9 Btu per pound. The theoretical compressor work is $(92.0 - 81.9) 219.8/42.4 = 52.4$ hp.

Multiple-expansion valves and back-pressure valves may be used, as shown in Fig. 13.3. All the refrigerant flows from the condenser through the first expansion valve, where its pressure is reduced from 139 to 55.4

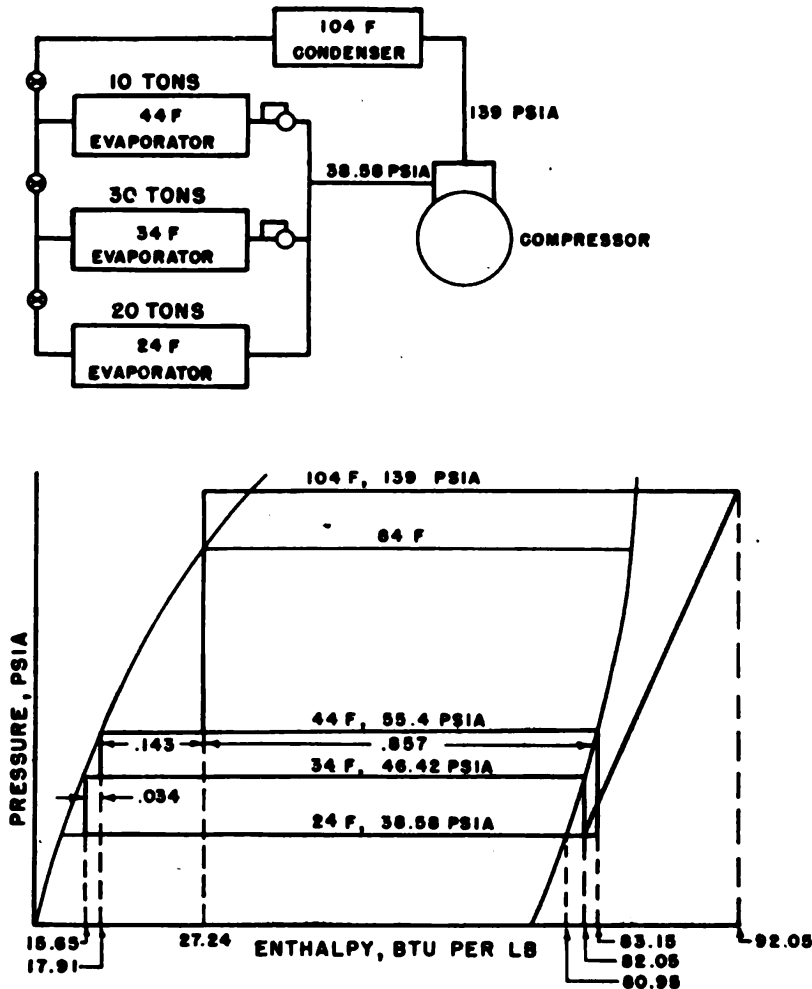


Fig. 13.3. Multiple expansion valves and back-pressure valves.

psia. All the vapor formed here plus enough liquid to take care of the 10-ton load flows through the 44 F evaporator. The remaining refrigerant then flows through the next expansion valve from 55.4 to 46.4 psia. All the vapor formed here plus enough liquid to take care of the 30-ton load flows through the 34 F evaporator. The remaining refrigerant then flows through the third expansion valve and supplies the 20-ton load of the 24 F evaporator. As in the second system, back-pressure valves are installed in the suction lines from the two higher-temperature evaporators in order to maintain their higher coil pressures. All three suction lines are connected to a common 38.6-psia suction line leading to the compressor.

The refrigerating effect in the lowest temperature evaporator is (h_g at 24 F) - (h_f at 34 F) = $80.95 - 15.65 = 65.30$ Btu per pound. The rate of refrigerant flow required is $(20 \times 200)/65.3 = 60.1$ lb per minute.

The refrigerating effect in the 34 F evaporator is (h_g at 34 F) - (h_f at 44 F) = $82.05 - 17.91 = 64.14$ Btu per pound. The quantity of refrigerant needed is $(30 \times 200)/64.14 = 93.5$ lb per minute. In addition, the vapor formed when the 60.1 lb per minute for the 24 F evaporator expanded from 55.4 to 46.4 psi passes through the 34 F evaporator. Since the quality of the mixture leaving the second expansion valve is 0.034, this additional vapor is $[0.034/(1 - 0.034)] 60.1 = 2.1$ lb per minute. Therefore the total refrigerant flowing through the 34 F evaporator is $93.5 + 2.1 = 95.6$ lb per minute.

The refrigerating effect in the 44 F evaporator is (h_g at 44 F) - [h_f at 84 F, (104 F - 20 F)] = $83.15 - 27.24 = 55.91$ Btu per pound. The quantity of refrigerant needed is $(10 \times 200)/55.91 = 35.8$ lb per minute. The additional vapor flowing through the evaporator without cooling effect is equal to $[0.143/(1 - 0.143)] (95.6 + 60.1) = 26.0$ lb per minute. The total refrigerant flowing through the 44 F evaporator is $35.8 + 26.0 = 61.8$ lb per minute.

The total refrigerant used is $60.1 + 95.6 + 61.8 = 217.5$ lb per minute. The enthalpy of refrigerant mixture entering the compressor is $(60.1 \times 80.95 + 95.6 \times 82.05 + 61.8 \times 83.15)/217.5 = 82.15$ Btu per pound. The theoretical compressor work is $(92.05 - 82.05)217.5/42.4 = 51.3$ hp.

In the last two systems it is recommended that check valves be installed in the suction lines from each of the two higher-pressure evaporators.

13.3. Dual Compression. Dual-effect compressors have two suction inlets drawing vapor in from two different pressure conditions. The lower-pressure vapor enters the cylinder as the suction stroke starts. Near the end of the stroke the second inlet port is uncovered and the higher-pressure vapor enters, raising the cylinder pressure enough to close the low-pressure intake valve. After the suction stroke is completed, all the vapor in the cylinder is compressed and discharged. If all conditions remain constant, a given amount of vapor from each suction inlet will be compressed. A brief analysis of the power requirements will be given, patterned after the more detailed development presented by Sparks.²

It is assumed that the following items are known: the piston displacement, clearance, and per cent of stroke when the auxiliary port is opened in the compressor; the properties of the two entering vapors; the discharge pressure; and the pressure drop through each valve or port. Compression is assumed to be adiabatic.

² Sparks, N. R., *Theory of Mechanical Refrigeration*. New York: McGraw-Hill Book Company, Inc., 1938, Chapter VIII.

The lower pressure vapor is subjected to throttling through the valves, to mixing with the clearance gas, and to warming by contact with the cylinder walls. None of these effects is of great magnitude, and it will be assumed that the heating by the cylinder walls counteracts the cooling effect of the throttling and of the clearance-volume vapor. It is assumed, therefore, that the temperature of the vapor in the cylinder when the auxiliary port is opened is the same as that of the vapor in the low-pressure line.

The following symbols will be used:

Weight	= W	Internal energy	= u
Volume	= V	Specific volume	= v
Enthalpy	= h	Total pressure	= p
Work	= W_k	$\tau \dot{V}_s$	= A
Subscript 1	= lower suction pressure vapor		
Subscript 2	= higher suction pressure vapor		
Subscript 3	= discharge pressure vapor		
Subscript o	= cylinder condition when port is opened		
Subscript m	= cylinder condition when port is closed		
Subscript c	= cylinder clearance volume vapor		

The cylinder volumes V_o and V_m , which are the same, may be calculated from the given data. v_o may be determined from the given properties for the vapor at 1.

$$\text{Then} \quad W_c + W_1 = \frac{V_o}{v_o} \quad (13.1)$$

$$\text{and} \quad W_c + W_1 + W_2 = \frac{V_m}{v_m} \quad (13.2)$$

$$\text{or} \quad W_2 = \frac{V_m}{v_m} - (W_c + W_1) \quad (13.3)$$

If the admission of the higher-pressure vapor and the subsequent mixing are considered adiabatic, the following energy equation may be written:

$$(W_c + W_1)u_o + W_2h_2 = (W_c + W_1 + W_2)u_m \quad (13.4)$$

Equation 13.4 assumes that the vapor already in the cylinder, $W_c + W_1$, is static and contains internal energy only as compared with the incoming vapor W_2 which contains internal energy plus flow work. The latter, by definition, is numerically equal to the enthalpy of W_2 . The net input of energy by the piston as it moves from point o to point m is omitted in this development, since its effect is comparatively insignificant. It is equal to $CA \times (p_m - p_o) \times (\text{cylinder volume} - V_o)$, where C is the effective part of the "toe" of the $p - v$ diagram.

Substituting $(h_m - Ap_mv_m)$ for u_m in equation 13.4 gives

$$(W_c + W_1)u_o + W_2h_2 = [(W_c + W_1) + W_2](h_m - Ap_mv_m) \quad (13.5)$$

Substituting from equation 13.3 gives

$$\begin{aligned} (W_c + W_1)u_o + \left[\frac{V_m}{v_m} - W_c + W_1 \right] h_2 \\ = \left[(W_c + W_1) + \frac{V_m}{v_m} - (W_c + W_1) \right] (h_m - Ap_mv_m) \end{aligned} \quad (13.6)$$

Simplifying, this becomes

$$\frac{h_m - h_2}{v_m} = Ap_m - \frac{1}{V_m} (W_c + W_1)(h_2 - u_o) \quad (13.7)$$

Simplifying further, since $W_c + W_1 = W_o$ and $u_o = h_o - Ap_o v_o$, the right-hand side of equation 13.7 becomes

$$Ap_m - \frac{W_o}{V_o} h_2 - \frac{W_o}{v_o} h_o + \frac{W_o}{V_o} Ap_o v_o$$

but since

$$\frac{W_o}{V_o} = \frac{1}{v_o}$$

the expression becomes

$$Ap_m - \frac{h_2}{v_o} - \frac{h_o}{v_o} + \frac{Ap_o v_o}{v_o},$$

or

$$A(p_m - p_o) - \frac{(h_2 - h_o)}{V_o}$$

Since the properties at o are the same as at 1, and $p_m = p_2$,

$$\text{then} \quad \frac{h_m - h_2}{v_m} = A(p_2 - p_1) - \frac{h_2 - h_1}{v_1} \quad (13.8)$$

Since both h_m and v_m are unknown, it is suggested that a curve be plotted of the quantity $(h_m - h_2)/v_m$ for p_m , against temperature. Since the right-hand side of equation 13.8 can be evaluated, t_m can be determined from the curve, and thus the other properties at m may be found.

Since $W_c + W_1$ can be found from the data given for (V_o/v_o) , W_2 can now be determined from equation 13.3. The properties after isentropic compression, point 3, can be found now, so that $W_c = V_c/v_3$. Then W_1 , and thus $W_3 = W_1 + W_2$ may be calculated.

The theoretical power is

$$\text{hp} = \frac{(W_3 h_3 - W_1 h_1 - W_2 h_2 + Q)}{42.4} \quad (13.9)$$

and for adiabatic compression $Q = 0$.

Individual expansion valves and dual compression may be applied as shown in Fig. 13.4. Using equation 13.8 for the dual compressor connected to the two lowest-pressure evaporators results in

$$\begin{aligned}\frac{h_m - 82.05}{v_m} &= \frac{144}{778} (46.42 - 38.58) - \frac{82.05 - 80.95}{1.043} \\ &= \frac{144 \times 7.84}{778} - \frac{1.1}{1.043} = 0.40\end{aligned}$$

When $t_m = 36$ F, then $h_m = 82.4$ and $v_m = 0.88$, which satisfies the above

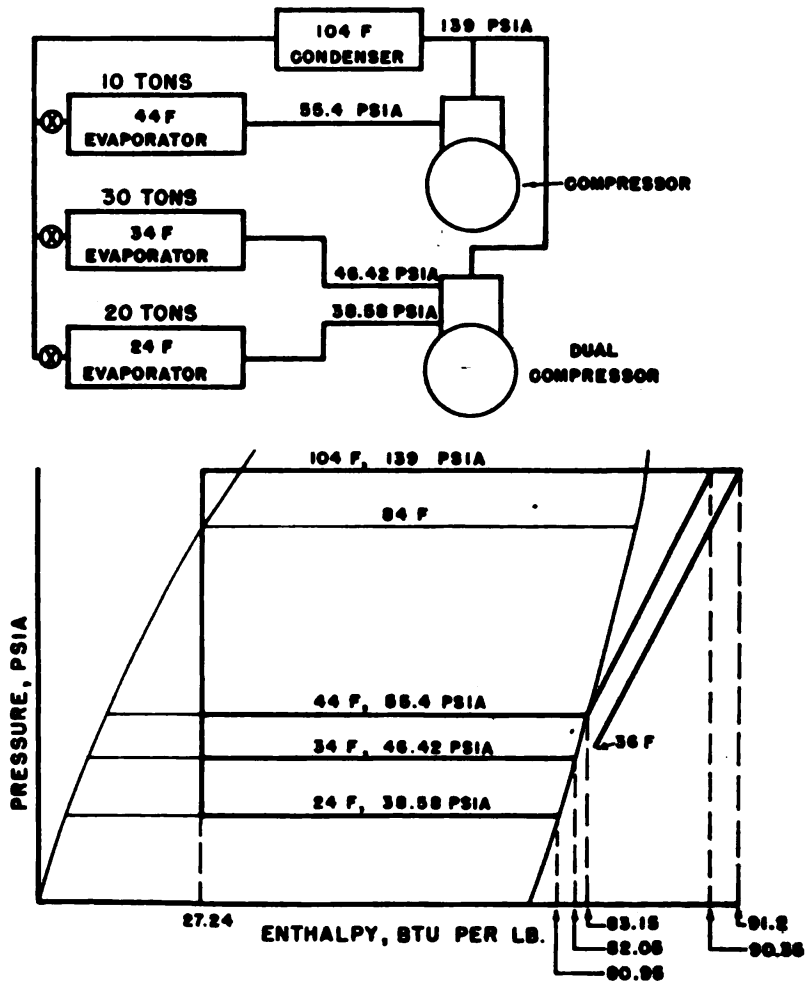


Fig. 13.4. Individual expansion valves and a dual compressor.

equation. Then

$$h_3 = 91.2$$

Weight of refrigerant in the 24 F evaporator:

$$W_1 = \frac{20 \times 200}{80.95 - 27.24} = 74.5 \text{ lb per min}$$

For the 34 F evaporator,

$$W_2 = \frac{30 \times 200}{82.05 - 27.24} = 109.5 \text{ lb per min}$$

The theoretical power for the dual compressor is

$$\frac{1}{42.4} [(74.5 + 109.5)91.2 - 74.5 \times 80.95 - 109.5 \times 82.05] = 41.6 \text{ hp}$$

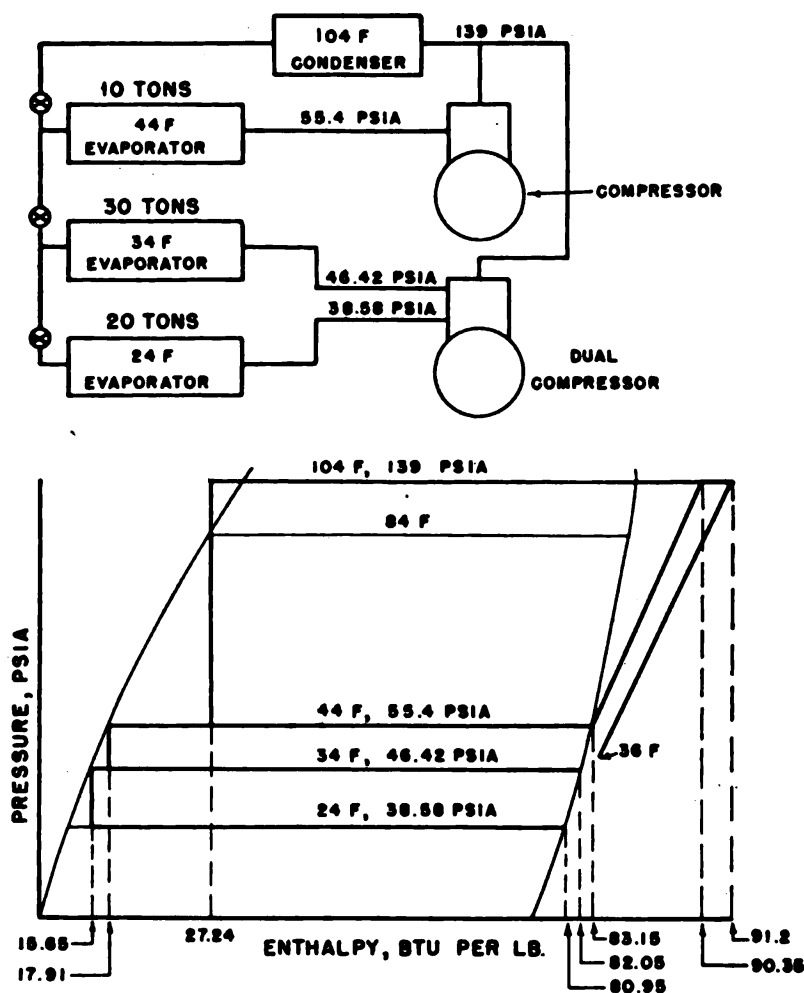


Fig. 13.5. Multiple expansion valves and a dual compressor.

The power for the other compressor is

$$\frac{10 \times 200}{83.15 - 27.24} \times \frac{(90.35 - 83.15)}{42.4} = 6.1 \text{ hp}$$

The total power is $41.6 + 6.1 = 47.7 \text{ hp}$.

Multiple expansion valves and a dual compressor would be connected as shown in Fig. 13.5. The method of analysis is the same as with individual expansion valves but the weights of refrigerant handled are different. As explained in §13.2,

$$W_1 = \frac{20 \times 200}{80.95 - 15.65} = 60.1 \text{ lb per min}$$

$$W_2 = \frac{30 \times 200}{82.05 - 17.91} + \frac{0.034 \times 60.1}{1 - 0.034} = 95.6 \text{ lb per min}$$

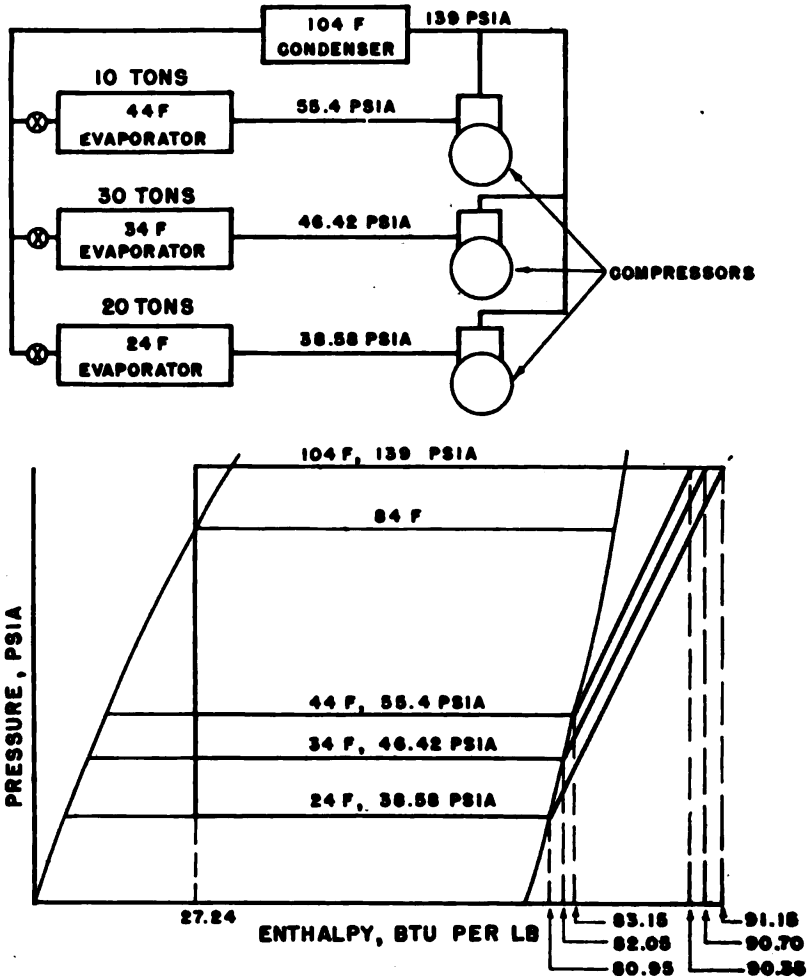


Fig. 13.6. Individual compressors and individual expansion valves.

The power for the dual compressor is therefore

$$\frac{1}{42.4} [(60.1 + 95.6)91.2 - 60.1 \times 80.95 - 95.6 \times 82.05] = 35.2 \text{ hp}$$

The power for the other compressor is

$$\left[\frac{10 \times 200}{83.15 - 27.24} + \frac{0.143}{1 - 0.143} (95.6 + 60.1) \right] \frac{(90.35 - 83.15)}{42.4} = 10.5 \text{ hp}$$

The total power is $35.2 + 10.5 = 45.7 \text{ hp}$.

13.4. Individual Compressors. In a completely sectionalized plant there is a separate compressor for each evaporator, and either separate condensers or one main condenser is used.

Using individual expansion valves, the system would be connected as shown in Fig. 13.6. The flow in the 24 F evaporator is $(20 \times 200)/(80.95$

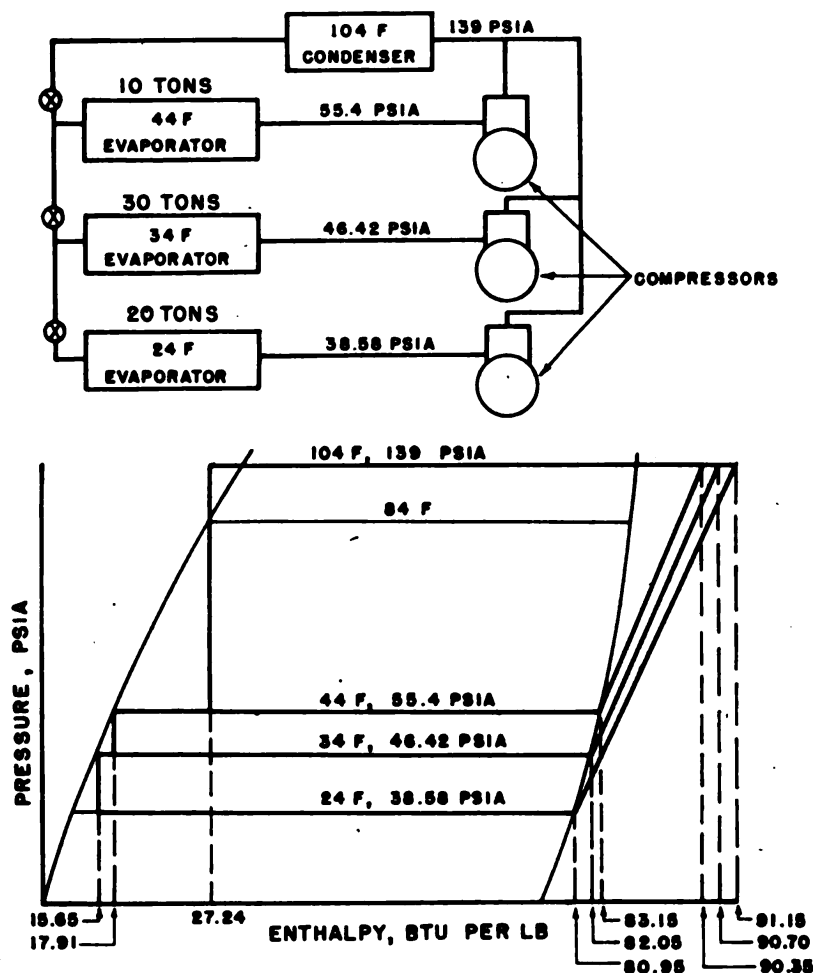


Fig. 13.7. Individual compressors and multiple expansion valves.

$- 27.24) = 74.5$ lb per minute. The power required is $74.5(91.15 - 80.95)/42.4 = 17.9$ hp.

In the 34 F evaporator the flow is $(30 \times 200)/(82.05 - 27.24) = 109.5$ lb per minute. The power required is $109.5(90.70 - 82.05)/42.4 = 22.4$ hp.

In the 44 F evaporator the flow is $(10 \times 200)/(83.15 - 27.24) = 35.8$ lb per minute. The power required is $35.8(90.35 - 83.15)/42.4 = 6.1$ hp.

The total power is $17.9 + 22.4 + 6.1 = 46.4$ hp.

Using multiple expansion valves necessitates using only one condenser, and the system would be connected as shown in Fig. 13.7.

In the 24 F evaporator the flow is $(20 \times 200)/(80.95 - 15.65) = 60.1$ lb per minute. The power required is $60.1(91.15 - 80.95)/42.4 = 14.5$ hp.

In the 34 F evaporator the flow is $(30 \times 200)/(82.05 - 17.91) = 93.5$ lb per minute plus the vapor formed when the 60.1 lb for the 24 F

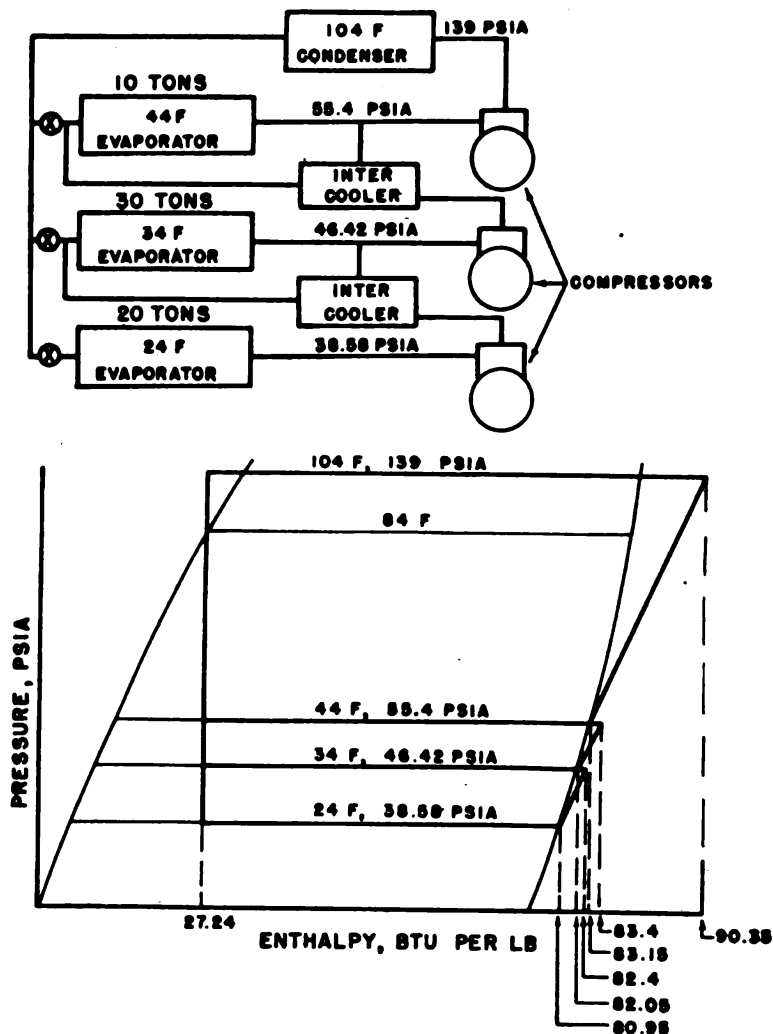


Fig. 13.8. Compound compression with flash intercoolers and individual expansion valves.

evaporator expands from 44 F to 34 F and a quality of 0.034. This additional vapor is $[0.034/(1 - 0.034)]60.1 = 2.1$ lb per minute. The power required is $(93.5 + 2.1)(90.70 - 82.05)/42.4 = 19.5$ hp.

In the 44 F evaporator the flow is $(10 \times 200)/(83.15 - 27.24) = 35.8$ lb per minute plus the vapor formed when the 60.1 lb and the 95.6 lb expand from 84 F to 44 F and a quality of 0.143. This additional vapor is $[0.143/(1 - 0.143)](60.1 + 95.6) = 26.0$ lb per minute. The power required is $61.8(90.35 - 83.15)/42.4 = 10.5$ hp.

The total power required is $14.5 + 19.5 + 10.5 = 44.5$ hp.

13.5. Compound Compression. Isothermal compression, requiring less power than isentropic compression, can be approached more closely with compound or stage compression. Intercooling may be done economically by allowing some of the refrigerant to by-pass the evaporators and be used in flash-type intercoolers.

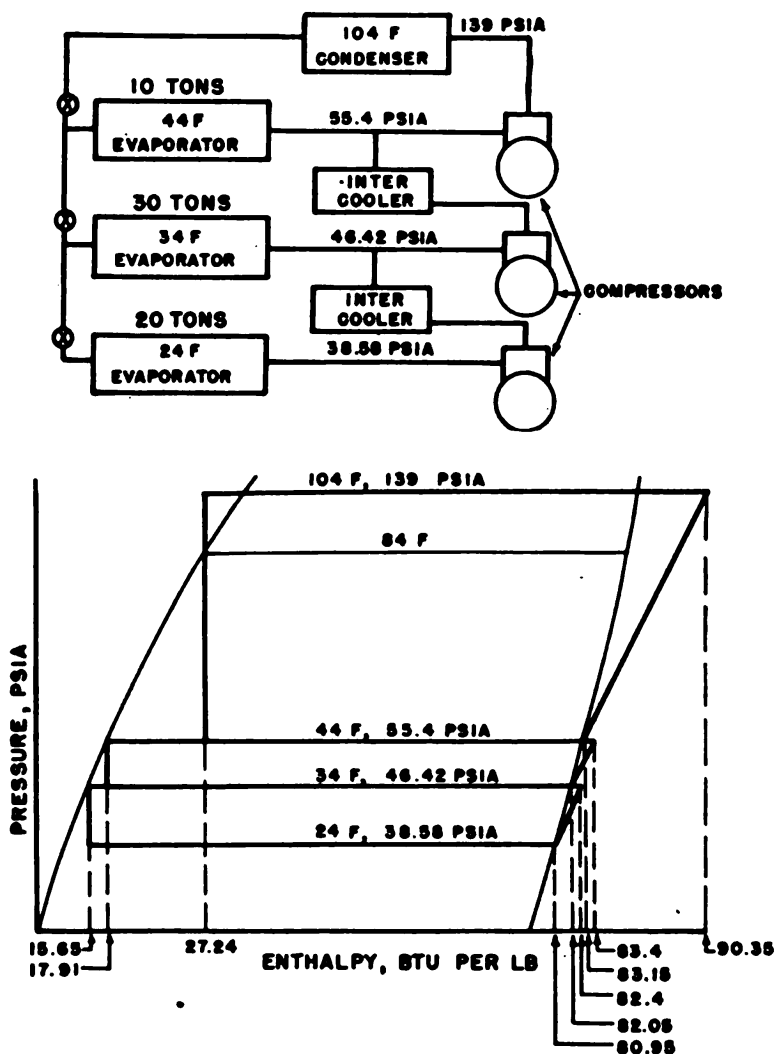


Fig. 13.9. Compound compression with flash intercoolers and multiple expansion valves.

Individual expansion valves with compound compression and flash intercoolers would be connected as shown in Fig. 13.8. The power for the low-pressure stage is $74.5(82.4 - 80.95)/42.4 = 2.55$ hp.

In the following intercooler the cooling effect required is $74.5(82.4 - 82.05) = 26.1$ Btu per minute. The amount of liquid refrigerant required is $26.1/(82.05 - 27.24) = 0.48$ lb per minute. The power for the middle stage compressor is $(74.5 + 0.5 + 109.5)(83.4 - 82.05)/42.4 = 5.87$ hp.

In the next intercooler the cooling effect required is $(184.5)(83.4 -$

83.15) = 46.1 Btu per minute. The amount of liquid refrigerant required is $46.1/(83.15 - 27.24) = 0.82$ lb per minute. The power for the final stage is $(184.5 + 0.8 + 35.8)(90.35 - 83.15)/42.4 = 37.6$ hp.

The total power required is $2.55 + 5.87 + 37.6 = 46.0$ hp.

Multiple expansion valves with compound compression and flash intercoolers are shown in Fig. 13.9. The power for the first stage is $60.1(82.4 - 80.95)/42.4 = 2.06$ hp.

TABLE 13.1
THEORETICAL COMPRESSOR POWER FOR A FREON-12 PLANT REQUIRING 10 TONS
AT 44 F, 30 TONS AT 34 F AND 20 TONS AT 24 F

Type of System	hp	% Reduction from Maximum
One compressor:		
All evaporators at same temperature.....	53.6	0.0
Individual exp. valves and back-pressure valves.....	52.4	2.2
Multiple exp. valves and back-pressure valves.....	51.3	4.3
Two compressors (one dual-effect type):		
Individual expansion valves.....	47.7	11.0
Multiple expansion valves.....	45.7	14.7
Three individual compressors:		
Individual expansion valves.....	46.4	13.4
Multiple expansion valves.....	44.5	17.0
Compound compressors and intercoolers:		
Individual expansion valves.....	46.0	14.2
Multiple expansion valves.....	44.2	17.5

In the first precooler the cooling effect required is $60.1(82.4 - 82.05) = 21$ Btu per minute. The liquid refrigerant required is $21/(82.05 - 17.91) = 0.33$ lb per minute. The power for the middle-stage compressor is $(60.1 + 0.3 + 93.5 + 2.1)(83.4 - 82.05)/42.4 = 4.96$ hp.

In the next intercooler the cooling effect required is $(156.0)(83.4 - 83.15) = 54.6$ Btu per minute. The liquid refrigerant required is $54.6/(83.15 - 27.24) = 0.98$ lb per minute. The power for the final stage is $(156.0 + 0.98 + 35.8 + 26.0)(90.35 - 83.15)/42.4 = 37.2$ hp.

The total power required is $2.06 + 4.96 + 37.2 = 44.2$ hp.

13.6. Comparisons of Systems. The theoretical compressor horsepower values calculated in the preceding sections for the cooling of three separate loads for a particular plant are listed in Table 13.1. In general, the total power required can be reduced appreciably by using more compressors or other equipment. The use of multiple expansion valves instead of individual expansion valves also reduces the power requirement somewhat. However, in certain multiload combinations this statement may not be true.

Several large restaurants, hotels, and institutions have recently converted their single-compressor ammonia, carbon dioxide, or methyl chlo-

ride multievaporator systems to multiple Freon-12 compressor systems. Economic studies have indicated that the saving in operating costs warranted the change-over. However, it generally is not feasible to go to the extreme of using stage compression and flash intercoolers on ordinary installations. These refinements are usually applied only to special industrial systems. The greater the difference between the evaporator temperatures required, the more practical is the use of multiple compressors. When the evaporators are near each other and when their design operating temperatures are not too different, a single compressor and back pressure valves would require the least space and may be the most economical arrangement. In any event, each problem should be completely analyzed in order to arrive at the most economical and feasible system. Control methods are described in §14.10.

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3. Sparks, N. R., *Theory of Mechanical Refrigeration*. New York: McGraw-Hill Book Company, Inc., 1938, Chapters VI and VIII.

PROBLEMS

13.1. Determine the percentage difference in theoretical power required (a) for F-12 and (b) for ammonia for one compressor with separate expansion valves and back-pressure valves and for individual compressors for evaporator loads of 5 tons at -10°F , 3 tons at 25°F , and 5 tons at 35°F ; condenser temperature is 90°F .

13.2. For an ammonia plant with loads of 10 tons at 44°F , 30 tons at 34°F , and 20 tons at 24°F , calculate and compare the power requirements for the three types of systems using one compressor as explained in §13.2. Condenser is at 104°F and there is 10°F subcooling.

13.3. An ammonia system supplies chilled brine for a 10-ton load at 0°F , a 2-ton load at 20°F , and a 4-ton load at 35°F and must operate at a -10°F suction temperature. Compare the theoretical power requirements with that for three separate F-12 compressors using direct expansion (no brine). Condensing temperature is 80°F with 10°F subcooling. Saturated vapor enters the compressor.

13.4. Determine the additional power saved in Problem 13.3 if multiple expansion valves are used in the F-12 system.

13.5. A 10-ton load at 20°F and a 10-ton load at 30°F are served by a single F-12 compressor with a back-pressure valve in the 30°F line. Compare the theoretical power required with that for two separate compressor systems. Condenser is at 100°F .

13.6. Make the same comparisons for the same loads as in Problem 13.5 but at 0°F and 30°F .

13.7. Determine the theoretical power requirements for a dual compressor for the conditions of Problem 13.5.

13.8. Determine the theoretical power requirements for 10 tons at 0 F and 10 tons at 30 F, using compound F-12 compression with an intercooler and multiple expansion valves and condenser at 100 F.

13.9. Work Problem 13.8, using ammonia.

13.10. Determine the power requirements for a system using individual compressors with multiple expansion valves for four 5-ton loads at 0, 10, 20, and 30 F evaporator temperatures, using Freon-12 and condensing at 90 F.

CHAPTER 14

Refrigeration Control

14.1. Introduction. For satisfactory operation, all but the largest refrigeration systems must be completely automatic. Large centrally operated systems requiring constant manual attendance are the exception. The overwhelming majority of installations are of small tonnage and are limited to a single application. Most of the usefulness of a domestic system or a small commercial unit would be lost if the operation were not fully automatic.

Automatic control of a refrigeration system requires complete control of the *flow of the liquid refrigerant*, the *on-off operation* of the compressor motor, the *flow of the condensing medium*, and *safety devices* for prevention of damage to equipment. In addition, special controls designed for specific applications are frequently required. The various types of devices used to accomplish these purposes are so numerous that it would be impossible to describe all their modifications. However, the general purposes and operating characteristics of each class will be discussed, and typical mechanisms used to accomplish these purposes will be reviewed.

14.2. Liquid-Refrigerant Control—Automatic Expansion Valve. Some form of expansion device must be used to control the rate of flow of the liquid refrigerant between the high- and low-pressure sides of the system. Such a device is termed an *expansion valve* and may be controlled either manually or automatically. With few exceptions the manual or hand-operated valves are obsolete and are not applied to modern systems.

Thermodynamically, the expansion process is an irreversible adiabatic one (see Chapter 4). However, if refrigeration systems of much larger size than now normally employed were used, it would be economically feasible to install expansion engines to recover some of the work of expansion and at the same time to accomplish the pressure reduction. Such a process then would approach the isentropic more closely.

An *automatic or pressure-control expansion valve* operates to maintain a constant pressure in the evaporator. A typical expansion control of this type is shown in Fig. 14.1. The liquid refrigerant enters at *A* from the receiver, passes through the strainer *B*, is throttled through the orifice *C*, with opening size controlled by the needle *D*, and is discharged into the evaporator through the refrigerant outlet *E*. The needle valve *F* is connected by means of a yoke to a flexible bellows *G*, which expands or contracts with variations in the evaporator pressure communicated

through the refrigerant outlet. The position of the needle is also controlled by the degree of compression in the spring connected to the yoke and adjusted by the hand screw *H*. These two forces act to maintain a constant pressure in the evaporator by increasing or decreasing the flow of liquid refrigerant as the pressure falls or rises in the evaporator.

This type of expansion valve is applied to dry evaporators of the direct-expansion type. In operation, the control feeds a sufficient amount of liquid refrigerant to the coils to keep the evaporator pressure constant. If the compressor is stopped, the remaining liquid evaporates and the evaporator temperature and pressure increase, in turn closing the needle valve on the expansion control and thus stopping any further flow of refrigerant from the receiver to the evaporator. When the compressor is started again, the evaporator pressure decreases until a balance point is reached at which the expansion valve opens, feeding additional liquid refrigerant into the evaporator. Initially, evaporation takes place in the first part of the evaporator coil only until this portion of the coil is cooled by the evaporating refrigerant. Eventually the rate of heat transfer through the first portion of the coil is reduced, in turn allowing the evaporating refrigerant to travel progressively closer to the suction connection on the coil. The evaporator and compressor must be proportioned so as to prevent liquid from passing from the evaporator coil into the suction line under conditions of load equilibrium. To obviate such a difficulty this type of system is frequently designed with an excessive length of expansion coil.

14.3. Liquid-Refrigerant Control—Thermostatic Expansion Valve.

A *thermostatic expansion valve* controls the flow of liquid refrigerant to the evaporator in such a way as to keep practically the entire coil filled with evaporating refrigerant and also to keep a constant superheat in the refrigerant vapor leaving the coil. A thermostatic expansion valve is like an automatic expansion valve in construction but incorporates in addition a power element responsive to changes in the degree of superheat of the refrigerant gas leaving the coil. The typical thermostatic valve shown in Fig. 14.2 is similar to the automatic expansion valve of Fig. 14.1 but with the addition of the power element *A*. This power element consists of a bellows *B* connected by means of a capillary tube *C* to a feeler bulb *D*. The feeler bulb is normally strapped to the suction line at the discharge of the evaporator. The feeler bulb, capillary tube, and power-element bellows may be charged with either a liquid or a gas. The pressure exerted by the power element through the push rod *E* to the yoke and needle *F* increases with expansion of the charging liquid or gas. Thus with this type of expansion valve a starved condition in the evaporator will result in a greater superheat in the gas leaving the evaporator, and this superheated gas in turn will operate through the power element to increase the flow of liquid refrigerant. A flooded evaporator will reduce

the superheat at the discharge of the evaporator, and hence the flow of liquid refrigerant will tend to be reduced. Such a valve is usually adjusted to allow evaporating liquid refrigerant to travel within a few feet of the suction-line connection. In this way full advantage is taken of the entire evaporator surface, and the system automatically adjusts itself to variations in load. Since no attempt is made to control the suc-

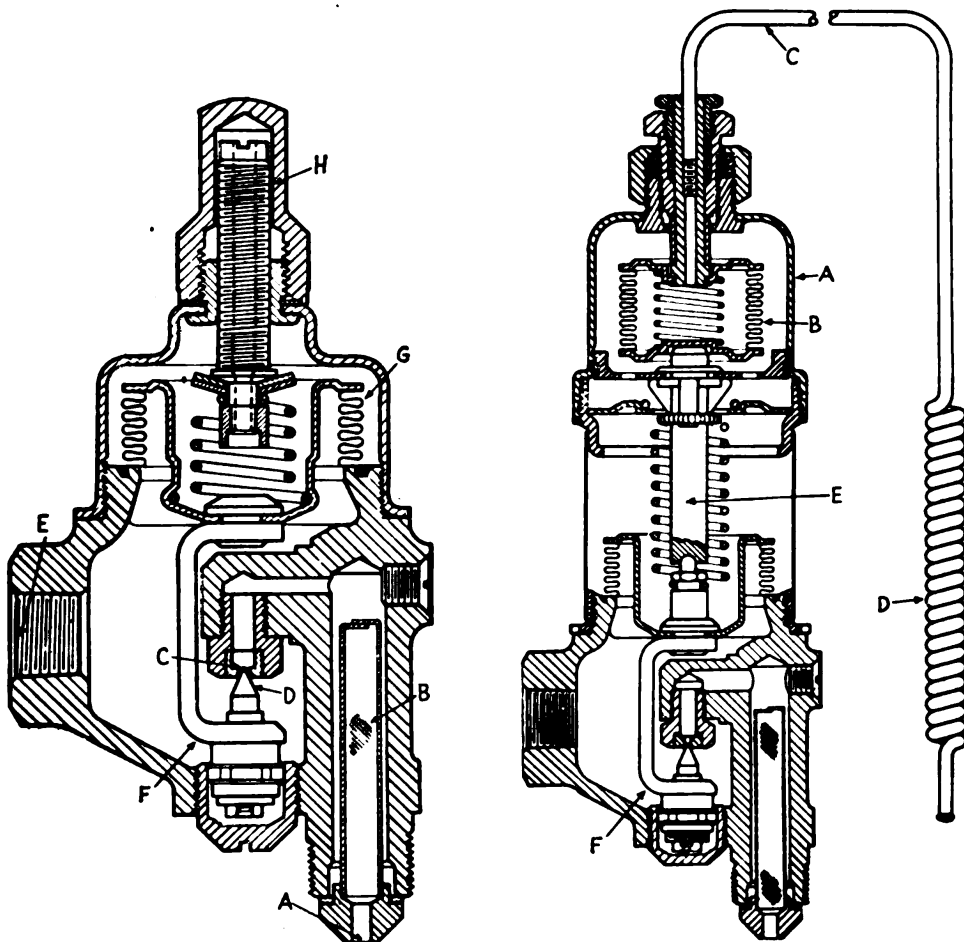


Fig. 14.1. Automatic expansion valve. Courtesy Detroit Lubricator Co.

Fig. 14.2. Thermostatic expansion valve. Courtesy Detroit Lubricator Co.

tion pressure, an auxiliary pressure control for starting and stopping the compressor motor may be used (see §14.7). Another design of thermostatic expansion valve is shown in Fig. 14.3.

If a thermostatic expansion-valve power element is charged with a liquid, the charging medium is usually the same refrigerant as used in the refrigerating system. Under these conditions the forces involved in positioning the needle of a thermostatic expansion valve are shown diagrammatically in Fig. 14.4. Here the refrigerant gas is Freon-12, the evaporator temperature 20 F, and the superheat of the suction vapor leaving the evaporator 10 F. The position of the expansion-valve needle

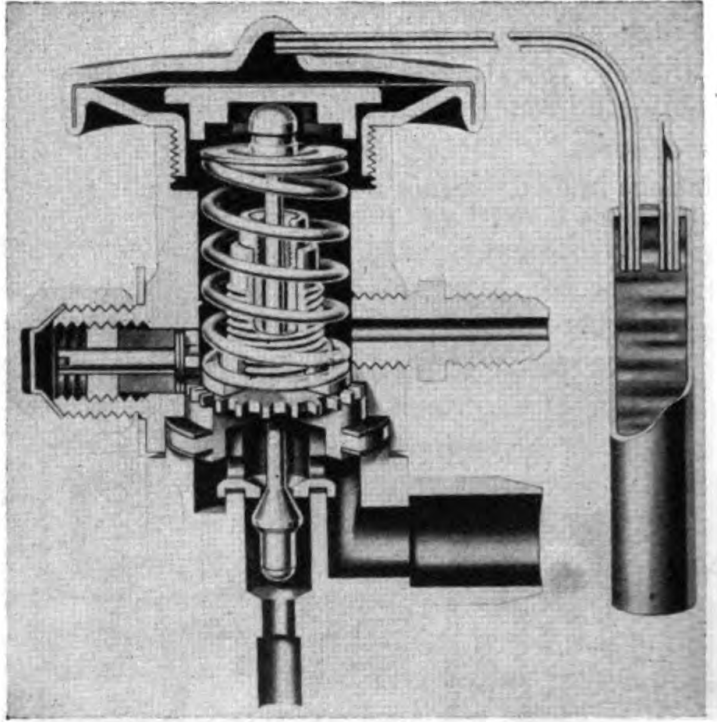


Fig. 14.3. Thermostatic expansion valve. Courtesy Alco Valve Co.

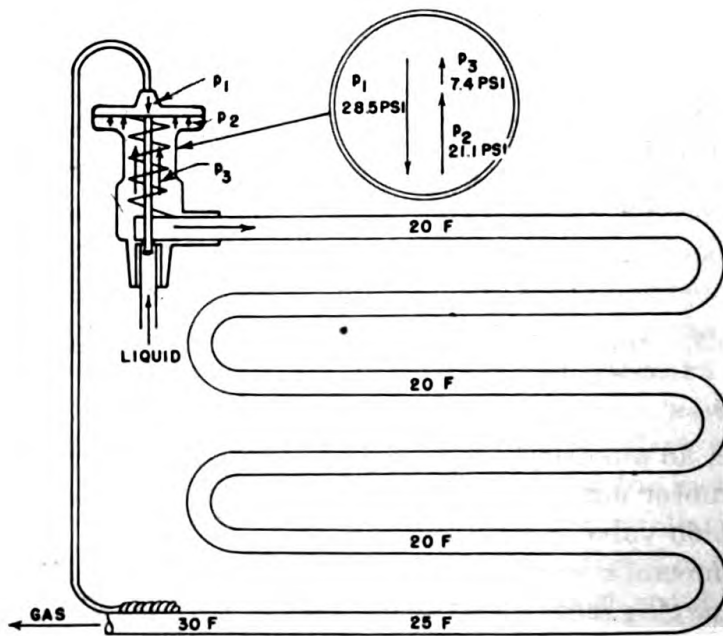


Fig. 14.4. Forces involved in operation of thermostatic expansion valve.

is determined by the three forces p_1 , p_2 , and p_3 which must balance to result in an equilibrium setting. The feeler-bulb pressure, p_1 , of 28.5 psi corresponds to the saturation pressure for Freon-12 at 30 F. This is balanced partially by force $p_2 = 21.1$ psi exerted against the diaphragm or bellows by the refrigerant at the entrance to the evaporator coil and by the pressure $p_3 = 7.4$ psi exerted by the restraining spring. If the

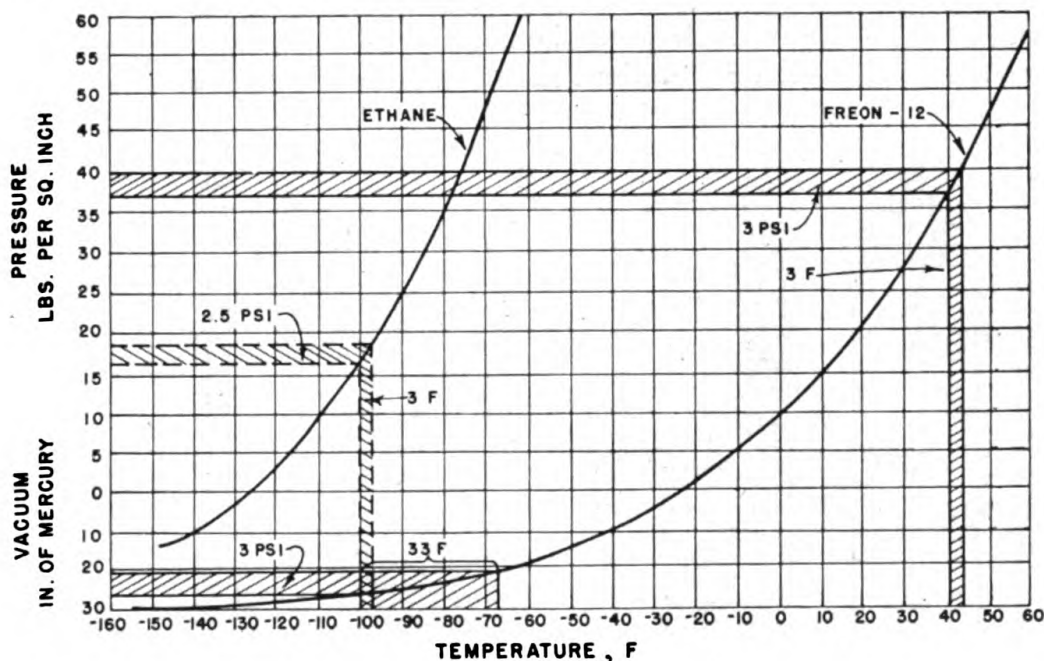


Fig. 14.5. Saturation curves for ethane and Freon-12. From "A Differential-Temperature Expansion Valve for Low-Temperature Applications," by F. Y. Carter. *Refrigerating Engineering*, Vol. 50 No. 1 (July, 1945).

cooling load increases, resulting in a superheat greater than 10 F at the discharge of the evaporator, pressure p_1 increases and thereby repositions the needle valve opening to allow a greater amount of liquid refrigerant to enter the evaporator. If the evaporator load decreases and thereby reduces the superheat below 10 F, force p_1 is reduced, and the needle valve closes to a position that reestablishes the 10 F superheat at the evaporator discharge.

A thermostatic expansion valve designed for comparatively high evaporator-temperature operation is unsatisfactory for extremely low-temperature applications, primarily because the pressure differential that must be supplied by the power element to change the needle valve from the open to closed position corresponds to a small superheat at high temperatures but to a very large superheat at low temperatures. This comparison is shown graphically in Fig. 14.5, which presents the saturation curves for Freon-12 and for ethane. If the refrigerant to be used is Freon-12 and if the power element of the thermostatic expansion valve is charged with Freon-12, a variation in temperature from 43 to 40 F will

result in a pressure change in the power element of from 40 to 37 psi. Thus a 3 F change in temperature corresponds to a 3-psi pressure change. However, at -100 F a 3-psi pressure differential corresponds to a 33 F change in superheat. Operation of such a thermostatic expansion valve at these low temperatures will therefore show wide superheat fluctuations between the open and closed positions of the valve if the system is to operate upon a 3-psi pressure differential and still supply the desired amount of refrigerant. It would be necessary to oversize the expansion valve greatly if such fluctuations are to be avoided; in this case a valve

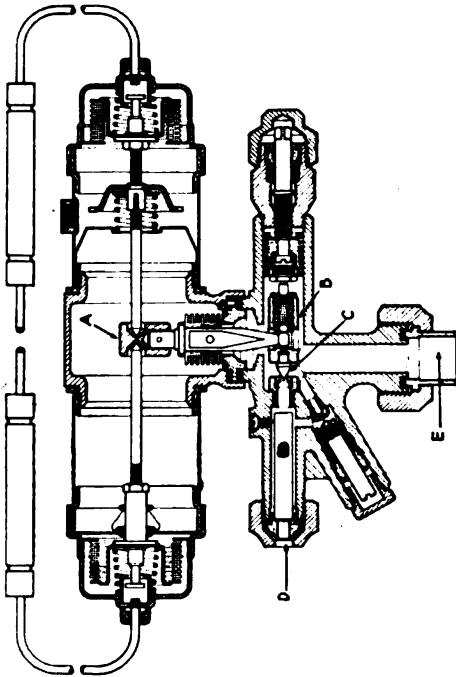


Fig. 14.6. Differential-temperature expansion valve. Courtesy Detroit Lubricator Co.

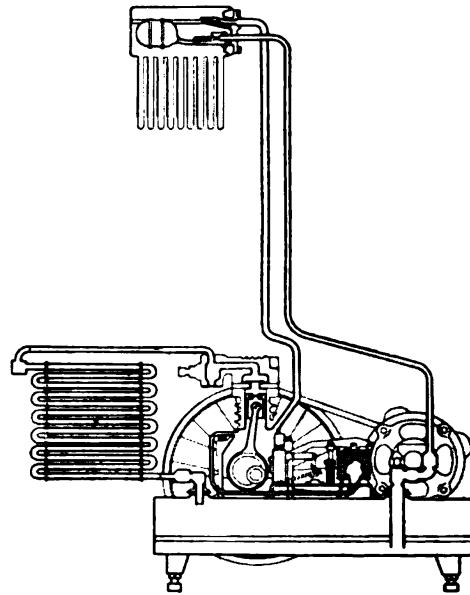


Fig. 14.7. Low-side float valve with flooded evaporator. Courtesy Frigidaire Division, General Motors Corp.

rated at 15 tons would be equivalent to a 1-ton valve at high temperatures. Several different schemes have been advanced to overcome these operating difficulties encountered at low temperatures. One such valve designed for low temperatures, shown in Fig. 14.6, operates upon a temperature-differential principle with two feeler bulbs, one to be attached to the discharge side of the evaporator and the other to the suction side. The feeler bulbs are charged with ethane, a refrigerant that exhibits a 2.5-psi pressure change with a 3 F change in superheat at -100 F. In this case small superheat changes result in pressure changes sufficiently great to be transmitted through operating lever *A* to the needle-valve assembly *B* to open and close the main needle valve *C* and thereby control the flow of liquid refrigerant from *D* through the outlet passage *E* to the evaporator.

14.4. Liquid-Refrigerant Control—Low-Side Float Valve. A liquid-refrigerant control of the *low-side float* type consists of a ball-type float located in either a chamber or the evaporator itself on the low-pressure side of the system and controlling a needle valve through a simple lever mechanism, as shown in Fig. 14.7 in cutaway view. This method of control is restricted to a flooded system and operates to maintain a constant level of the ebullient refrigerant in the evaporator. Such systems are used on household refrigerators and to some extent in commercial and industrial applications.

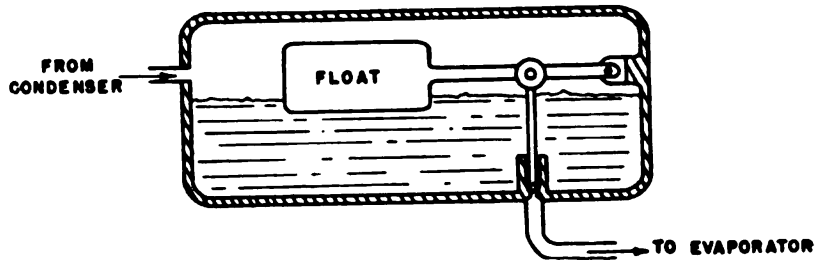


Fig. 14.8. High-side float valve.

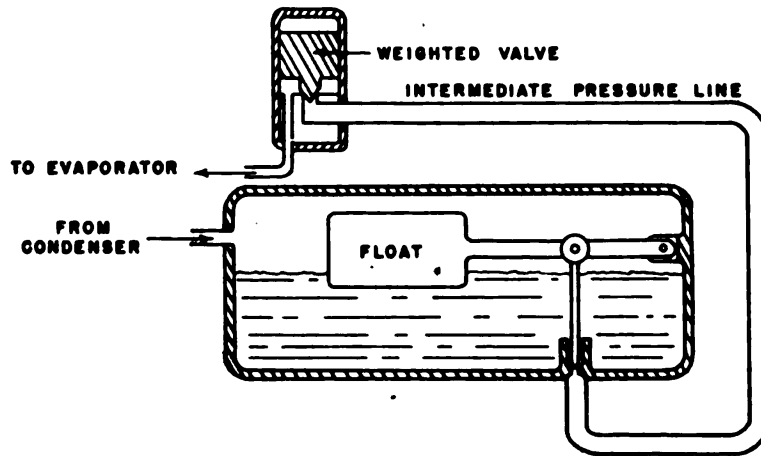


Fig. 14.9. High-side float valve with intermediate pressure valve.

14.5. Liquid-Refrigerant Control—High-Side Float Valve. A *high-side float* refrigerant control differs from a low-side float expansion valve in that the float itself is located in a liquid receiver or an auxiliary liquid container on the high-pressure side of the system. The liquid refrigerant is expanded either directly from this high-pressure chamber into the low-pressure side of the system or into an intermediate-pressure feed line. A high-side system of the first type is shown in Fig. 14.8 and one of the second type in Fig. 14.9. In the latter the liquid refrigerant passes from the high-pressure receiver into an intermediate-pressure feed line and thence through a second reducing valve, consisting of a fixed weight load acting upon a needle valve, into the evaporator. This system is designed with an intermediate pressure sufficiently high to reduce the heat gain

to this line to a minimum. In some instances this intermediate pressure line is replaced by a capillary-tube expansion system (see §14.6).

A high-side float expansion system of the design shown in Fig. 14.8 has the disadvantage that the evaporator must be placed directly adjacent to the float chamber containing the supply of high-pressure and high-temperature liquid refrigerant. The relatively high temperature differential results in a comparatively rapid transfer of heat from this reservoir

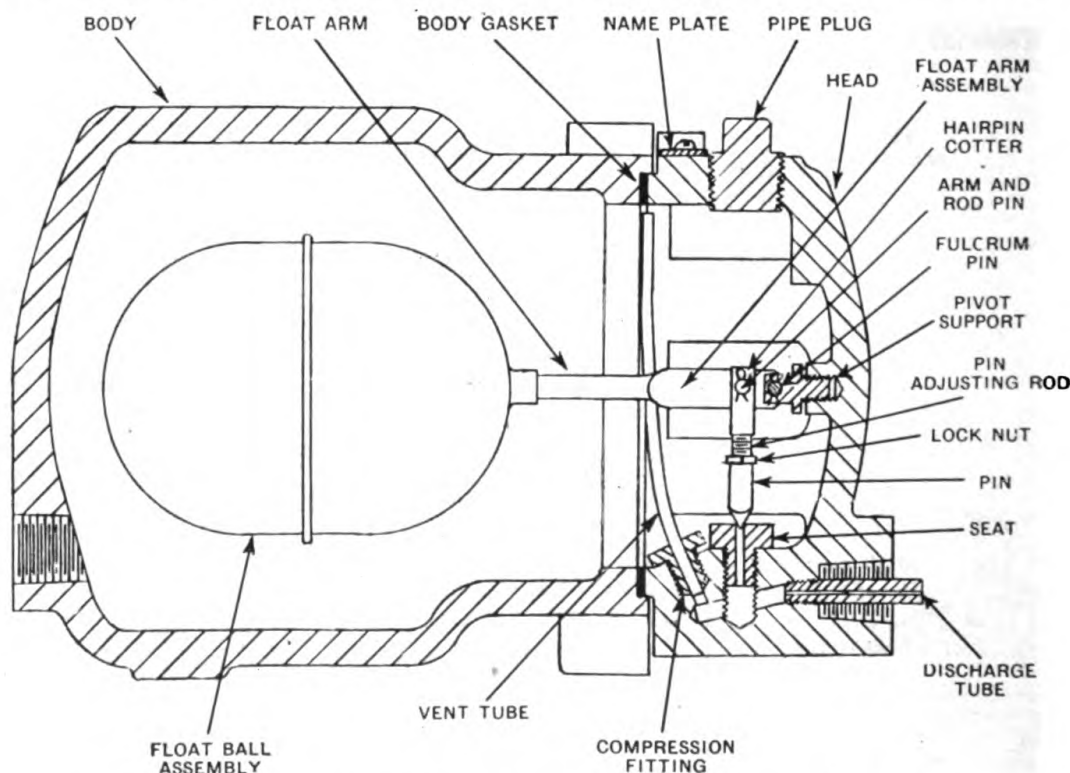


Fig. 14.10. Detail of high-side float assembly. Courtesy Alco Valve Co.

to the evaporating refrigerant with an accompanying loss of available refrigerating effect. For this reason the intermediate-pressure system or the combination high-side float and capillary system has found increasing usage, because in these, transfer of heat is reduced by separating the high-pressure chamber and the evaporating refrigerant by a reasonable distance. In either case the charge of liquid refrigerant in the system is critical if proper balance and operation are to be obtained. A typical high-side float assembly as used in domestic units is shown in Fig. 14.10.

14.6. Liquid-Refrigerant Control—Capillary Tube. A *capillary tube* when used as a liquid-refrigerant expanding device usually consists of an extremely small-bore tube (in the order of 0.042 in. diameter) of about 5 to 10 ft in length. Although such a device is a very simple means of expanding liquid refrigerant from high to low pressure, once the size and length of the tube are fixed there are no modifications possible to adjust for various operating conditions. The bore and length of the tube, there-

fore, as well as the proportions of the rest of the system, are critical, and for this reason its application has been limited to "packaged" domestic and commercial units.

When the liquid refrigerant passes through a capillary tube, the pressure is reduced gradually to the flash point at which a proportion of the liquid changes to gas. The proportion of the liquid changing state may be reduced, and therefore the capacity of the capillary tube increased, by utilizing a heat exchanger to subcool the entering liquid refrigerant. Such a heat exchanger may be formed by attaching the capillary tube to

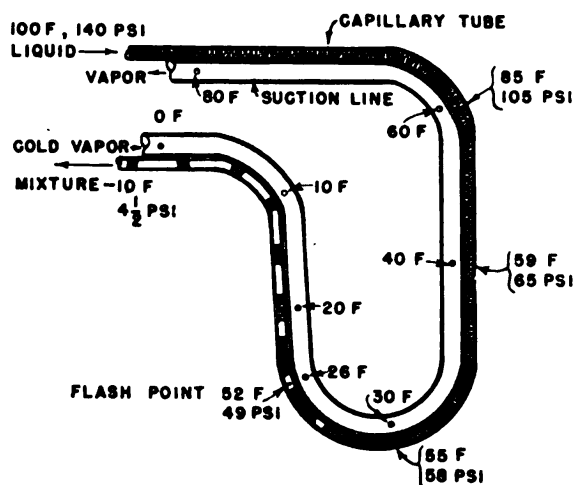


Fig. 14.11. Temperature-pressure distribution through typical capillary heat exchanger. From "Capillary Tube-Heat Exchangers," by R. H. Swart. *Refrigerating Engineering*, Vol. 52, No. 3 (September, 1946).

a length of the suction line returning the refrigerant gas from the evaporator to the compressor. Thus the cool gas leaving the evaporator may be used to subcool the warm liquid refrigerant leaving the condenser. The refrigerant liquid and vapor pressures and temperatures for such a capillary-tube heat exchanger are shown in Fig. 14.11 for an application using F-12 as the refrigerant. Here the cooled vapor returns from the evaporator at 0 F and is superheated to 80 F in the suction line before entering the compressor. At the same time the liquid refrigerant leaves the condenser at 100 F and 140 psi, is cooled to a temperature of 52 F before the pressure drop is sufficient to reach the saturated condition, and finally leaves the capillary tube to enter the evaporator at -10 F and 4.5 psi pressure. Without such a heat exchanger the refrigerant would start to flash to vapor shortly after entering the capillary tube, and therefore the capacity of the tube as well as the refrigerating effect of the refrigerant entering the evaporator would be greatly reduced.

The proper application of capillary tubing as an expansion device usually requires the use of a hermetic compressor. For satisfactory oper-

ation a capillary tube system requires the continuous maintenance of a fixed refrigerant charge, high standards of cleanliness and dehydration, and a uniform rate of flow of oil with the refrigerant.¹

A hermetic compressor inherently possesses these requirements to a greater degree than does the open type of compressor, subject to field servicing and to some loss of refrigerant past the compressor shaft seal. Proper balancing of the system parts is of utmost importance for proper operation. A capillary tube, with fixed dimensions and fixed resistance to flow, increases in ability to supply liquid refrigerant as the condenser pressure increases and/or the evaporator pressure decreases and the pressure differential thus becomes greater. Conversely, the ability of the remainder of the system to supply liquid refrigerant to the capillary tube decreases as the evaporator pressure drops and/or the condensing pressure increases. Thus there is a point termed "capacity balance" at which the ability of the condensing unit to supply liquid refrigerant is just matched by the flow of the refrigerant through the capillary tube and with a liquid seal at the entrance to the capillary. On either side of this point of capacity balance there will exist conditions under which either refrigerant vapor enters the capillary tube along with the liquid refrigerant or liquid refrigerant will build up in the condenser. Only one point of capacity balance will exist for any one set of operating conditions.

The principal advantage in the use of a capillary tube as an expansion device is its simplicity. There are no moving parts, and the system automatically unloads, with the high-side and the low-side pressures equalizing during periods when the compressor is not operating. This automatic action eliminates the need of any unloading device and permits the use of a motor with low starting torque. Somewhat less refrigerant is required in such a system, since there is no continuous storage of liquid under high pressure in the condenser during the period of "off cycle." Thus manufacturing costs are appreciably lower for systems using capillary tubes. However, because of the fixed bore and length of the tube and the varying operating conditions encountered, there is usually some increase in the average operating cost. Furthermore, extreme care must be taken to prevent plugging of the tube by any dirt, ice, or decomposition products.

14.7. Compressor Motor Controls. Two types of controls are commonly used for intermittent starting and stopping of compressors. The first of these is a *pressure motor control* responsive to the suction pressure, and the second is a *thermostatic motor control* responsive to the temperature of the load surrounding the evaporator. In the first, compressor operation is indirectly dependent upon the temperature of the load and

¹ Staebler, L. A., "Theory and Use of a Capillary Tube for Liquid Refrigerant Control." *Refrigerating Engineering*, Vol. 55, No. 1 (January, 1948), p. 55.

is controlled by the pressure, and therefore also by the temperature, of the evaporating refrigerant. In the second type, compressor operation is directly dependent upon the temperature of the load being cooled.

With the pressure-actuated device, the control, for convenience, is located directly on the condensing unit, and the low pressure in the suction line or the crankcase of the compressor is used to control motor operation.

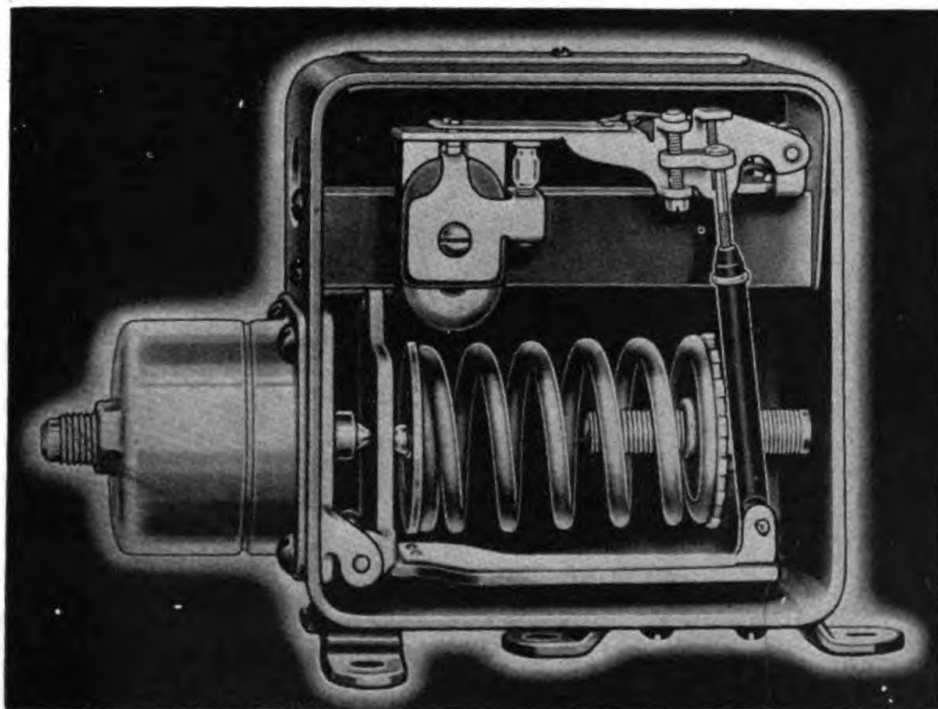


Fig. 14.12. Pressure motor control. Courtesy Penn Electric Switch Co.

As shown in Fig. 14.12, such a control consists usually of a low-pressure bellows connected through tubing directly to the low-pressure control source and an electrical switch operated through linkage by the movement of the bellows. The electrical circuit is closed on rising pressure and is opened on falling pressure. Adjustment may be made of the cut-in and cutout pressures as well as the differential between them. Usually the cutout pressure is adjusted to correspond to a temperature a few degrees below that of the desired coil temperature and the cut-in pressure approximately to that of the coil temperature. A wide pressure differential would tend to allow greater variation in cabinet temperature but would lengthen the cycling time. With a small pressure differential the cycling time will be shortened but the load temperature will be more uniform.

The thermostatic type of motor control is usually of similar construction to the pressure control, with the exception that a temperature bulb and capillary tube replace the pressure line. The bellows linkage and electrical contacts shown in the control of Fig. 14.12 may also be used for temperature control by equipping it with a temperature actuated power

element. In this case motor control is directly responsive to changes in the temperature of the load surrounding the evaporator if the power element is properly located. With this type of control, the cut-in temperature, the cutout temperature, and the differential are all adjustable.

Motor controls are frequently equipped with a high-pressure safety cutout switch that operates to cut off the power from the motor when the

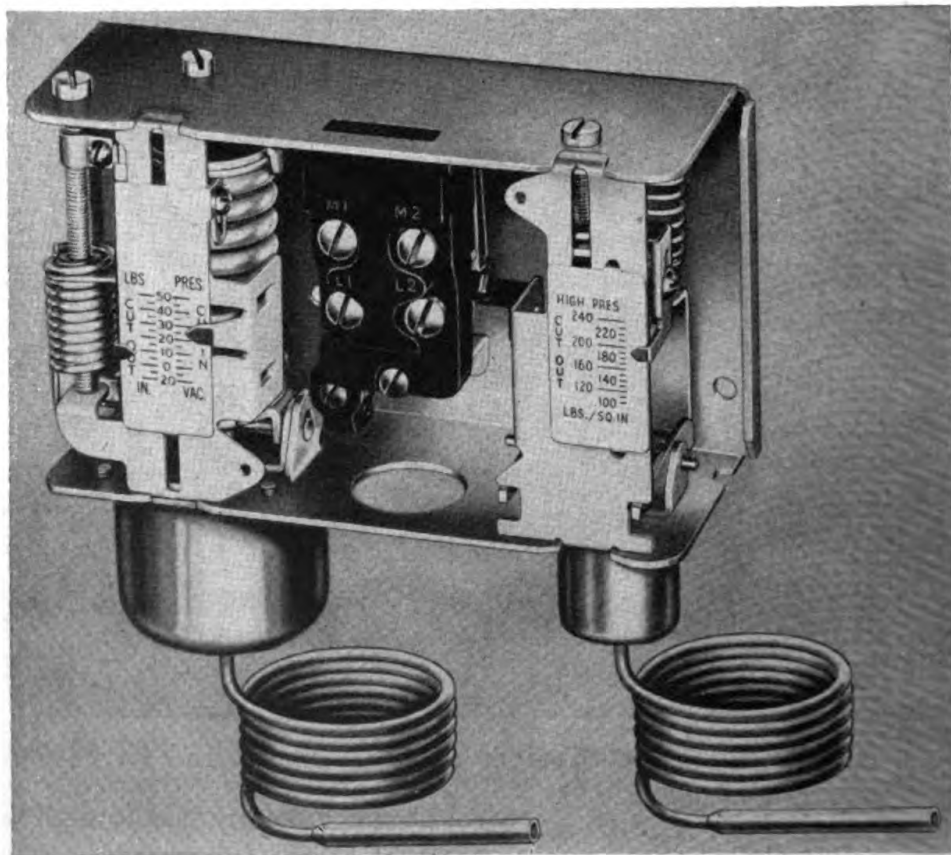


Fig. 14.13. Combination low-pressure control and high-pressure safety cutout. Courtesy Penn Electric Switch Co.

high-side pressure exceeds a predetermined limit. A typical combination low-pressure control and high-pressure safety cutout is shown in Fig. 14.13. The construction of the high-pressure cutout is similar to that of the low-pressure control in that it consists of a pressure-operated bellows connected by means of a linkage to an electrical switch. The cylinder head-pressure of the compressor is frequently used as the controlling point. If the pressure in the condenser exceeds the control limit through too little condenser water, an excess of noncondensable gases in the system, or any other cause, the system ceases to operate and damage is prevented. On large installations it is desirable that such a cutout be of the manual reset type.

14.8. Solenoid or Magnetic Stop Valves. *Solenoid or magnetic stop valves* are frequently used in refrigeration and air-conditioning systems

for control of gas or liquid flow. Because such a valve has two positions only (fully closed or fully open) and a fixed area when open, it cannot be used as a modulating control. A typical solenoid valve is shown in Fig. 14.14. If the solenoid coil is not energized, the plunger and needle close the orifice, preventing flow; when the solenoid coil is energized, the plunger is drawn into the solenoid, thus opening the orifice passage.

Solenoid valves are frequently used as liquid stop valves and are placed in the liquid line between the condenser or receiver and the evaporator. The liquid-refrigerant line is open to passage of the refrigerant only when the compressor is in operation and the solenoid is energized; hence leakage of liquid refrigerant into the evaporator is prevented when the compressor is inoperative. In some cases a magnetic liquid shut-off valve is operated directly by a thermostat located at the point of the load, and the compressor-motor operation is controlled independently by a low-pressure switch. Closing the liquid line through the thermostat permits the compressor to evacuate the evaporator, and the resulting low pressure stops compressor operation.

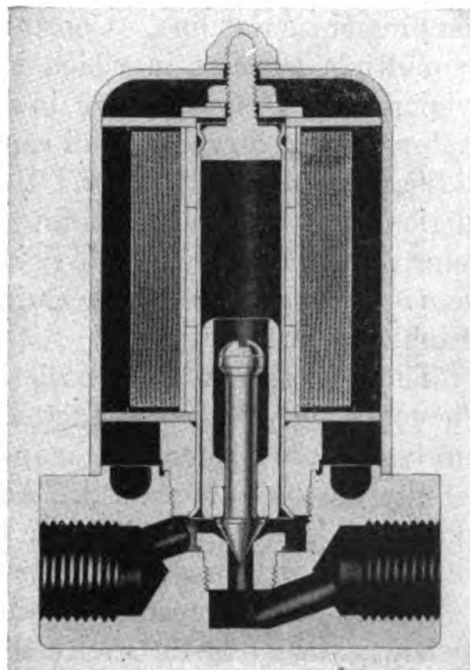


Fig. 14.14. Solenoid or magnetic stop valve. Courtesy Penn Electric Switch Co.

Magnetic stop valves are widely used for the control of refrigerant flow to individual evaporators in a multiple-evaporator system operated by one compressor. This phase of refrigeration control is more fully discussed in §14.10. Many other uses are found for magnetic valves of this type. In some installations magnetic liquid-line and suction-line valves are used to isolate the evaporator completely for defrosting purposes. A magnetic valve may be installed in a by-pass around one or more cylinders of a multiple-cylinder compressor and thereby be used to provide a degree of capacity control. A similar by-pass gas solenoid valve may be used to unload a compressor during starting to reduce the starting load. Additional applications are found in control of the flow of condenser water or in control of the circulation of brine in a secondary refrigeration system.

14.9. Suction-Line Controls. *Suction-pressure control* valves, frequently called *back-pressure regulators* or *two-temperature* valves, are placed in the suction line to prevent the evaporator pressure, and there-

fore the evaporator temperature, from dropping below a predetermined level. They may be applied to installations such as water-cooling and milk-cooling systems in which freezing or other damage may result if the evaporator pressure drops too low. They are often used in multiple systems in which several evaporators are supplied by one condensing unit. With such installations the different evaporators may be kept at different temperatures by maintaining a pressure drop from the evaporator to the compressor suction line. Constant-suction pressure valves also find usage in multiple systems in which individual control is desirable for each evaporator. If, for example, in a two-evaporator system one of the evaporators is shut down, the full capacity of the compressor is then applied to the remaining evaporator. Unless a device of this type were used, the suction pressure in the evaporator would then be reduced to a much lower value than that for which the system was designed when both evaporators were operating. Suction pressure control has, however, the disadvantage of inherent instability.

There are several different designs of constant-suction pressure valves. The metering type is constructed to open and close with only slight pressure variation, whereas the snap-action diaphragm type operates over a definite pressure differential. Additional modifications have been made so that back-pressure valves can be operated by means of a thermostatic control bulb in a manner similar to a thermostatic expansion valve.

One typical evaporator-pressure regulator of the pilot-operated type is shown in Fig. 14.15. With this pressure regulator the main valve is held in the closed position by the evaporator pressure and by the valve spring. The evaporator pressure is transmitted to the pilot valve *E* either through an internal or an external passage, depending upon the application. The pilot spring *C* is adjustable so that the pilot may be made to open whenever the evaporator pressure exceeds a predetermined setting and to close whenever the evaporator pressure drops below this value. If the pressure in the evaporator is sufficiently high to counteract the spring pressure in the pilot and to open the pilot, evaporator pressure is then transmitted through the pilot opening to the top of the main-valve operating piston *J*. Since the area of this piston is greater than that of the main valve, this pressure is sufficient to counteract the forces holding the main valve closed, thus opening the valve and allowing free passage from the evaporator to the suction line. In this case a 2-psi pressure differential between the suction line and the evaporator is sufficient to result in proper operation.

A comparatively recent development in suction-line controls is a suction-pressure regulator designed to limit the suction pressure at the compressor to a maximum value and thus to prevent the compressor from overloading, particularly during the initial high-load or "pulldown" periods. Some installations have been made combining such a suction-

pressure regulator with an evaporator-pressure regulator in the same system, thus limiting the compressor load during pulldown and controlling the evaporator pressure at a minimum value during normal and light loads. Such control systems, as well as other similar combinations using temperature-controlled suction pressure valves, make possible the safe selection of a compressor based upon the maximum necessary load and at the same time permit safe operation during periods of overload.

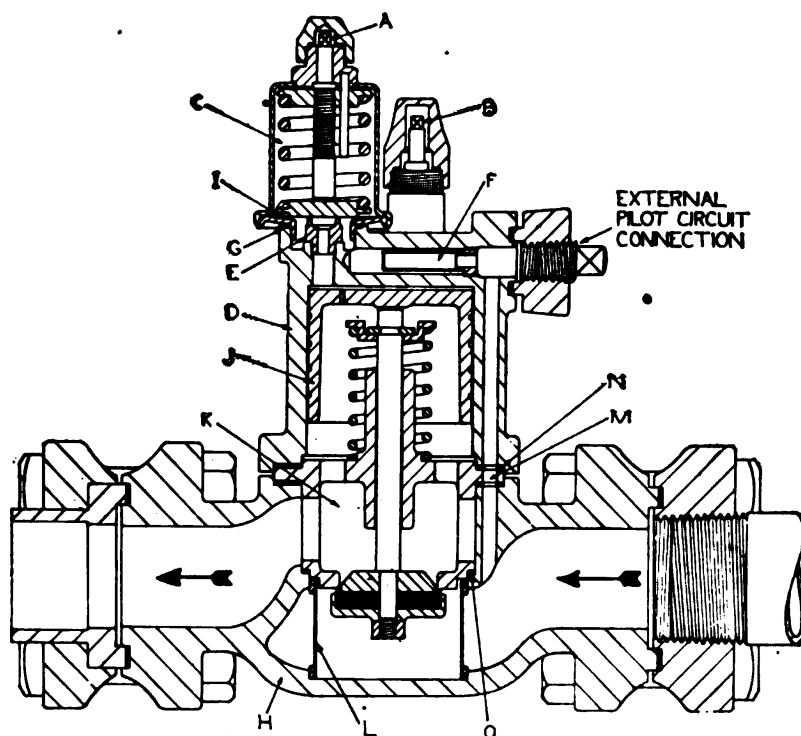


Fig. 14.15. Evaporator pressure regulator. Courtesy Alco Valve Co.

14.10. Control of Multiple-Evaporator Systems. If several evaporators are required for the removal of heat from different sources, these may be operated by means of individual condensing units for each evaporator or by one large condensing unit of sufficient capacity to supply the total of all the loads. For individual units the required controls are the same as for any individual system, but special controls are necessary for the system employing one large unit.

If the several evaporators are to be operated at the same temperature and the several loads are all equal, any single evaporator pressure or any single load temperature would be sufficient to control operation of the condensing unit. However, these conditions are obtained rarely, if ever, since not only will the loads vary somewhat but the desired evaporator temperatures also may not be the same. Different evaporator pressures may be maintained by the use of suction pressure-throttling devices (see §14.9). If proper temperature and load control are to be maintained for

each evaporator, some means must be provided for individual evaporator control by shutting off the refrigerant flow as soon as the desired load temperature has been reached.

One satisfactory means of providing multiple-evaporator control when operating with a single condensing unit is through the application of *thermostat-solenoid* combinations together with *suction-line pressure-throttling* devices. Such an installation, as shown diagrammatically in Fig. 14.16, is equipped with individual liquid-line solenoid valves thermostatically controlled by the load prevailing in the areas surrounding the

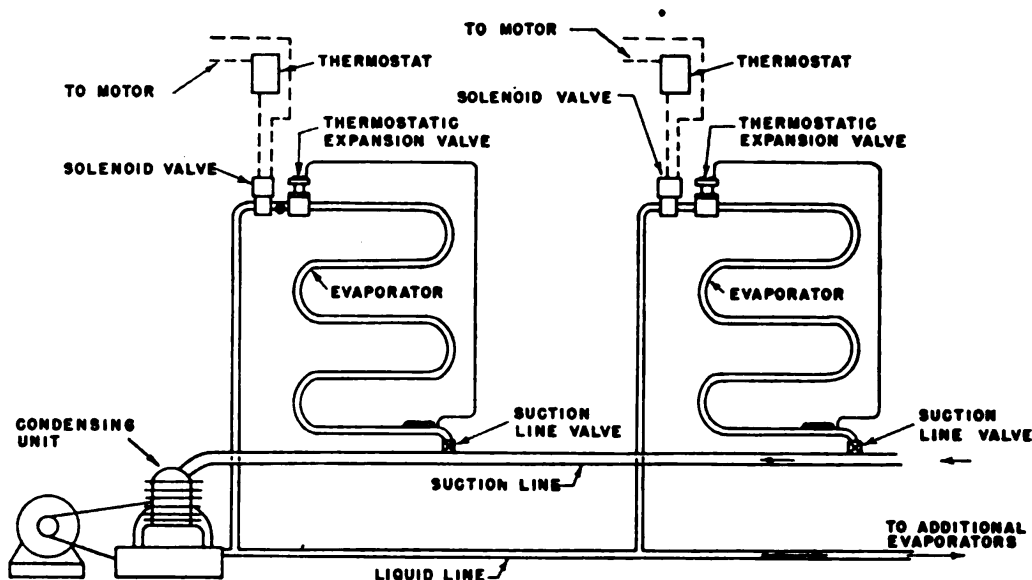


Fig. 14.16. Thermostat-solenoid control system.

evaporator. The compressor operates when any one or all thermostats request refrigeration. Expansion control is provided through thermostatic expansion valves, and pressure-restricting valves on the suction line of each evaporator prevent low evaporator temperatures when the full capacity of the compressor is supplied to less than the maximum number of evaporators in the system. A suction-line throttling device is particularly necessary if low evaporator pressures and temperatures might cause damage, such as dehydration in the case of food products.

A thermostat-solenoid system as here described gives individual load control for each evaporator and prevents low evaporator temperatures during partial-load operation. However, the operating cost is increased, not only because throttling is necessary between the suction line and those evaporators operating at higher pressures than the minimum but also because the full capacity of the condensing unit is supplied whether only one or all evaporators are demanding condensing unit operation. If it is assumed that t equals the total running time for the condensing unit, I equals the individual operating time in per cent for each evaporator,

and n equals the number of evaporators, then by application of the probability laws:²

Probability of condensing unit on and operating all evaporators at same time =

$$\frac{I_1}{100} \times \frac{I_2}{100} \times \frac{I_3}{100} \times \dots \times \frac{I_n}{100} = \frac{I^n}{100^n} \quad (14.1)$$

Probability of condensing unit off =

$$\begin{aligned} \frac{(100 - I_1)}{100} \times \frac{(100 - I_2)}{100} \times \frac{(100 - I_3)}{100} \times \dots \times \frac{(100 - I_n)}{100} \\ = \frac{(100 - I)^n}{100^n} \end{aligned} \quad (14.2)$$

Probability of condensing unit on and operating one or more but not all evaporators at same time =

$$1 - \left[\frac{I^n}{100^n} + \frac{(100 - I)^n}{100^n} \right] \quad (14.3)$$

Condensing unit total running time =

$$\begin{aligned} t &= \frac{I^n}{100^n} + \left[1 - \left(\frac{I^n}{100^n} + \frac{(100 - I)^n}{100^n} \right) \right] \\ &= 100 \left[\frac{- (100 - I)^n + 100^n}{100^n} \right] \end{aligned} \quad (14.4)$$

Similar equations may be derived for the more complex cases where the individual running times for each unit are not the same.

EXAMPLE 14.1. It is required that each of three evaporators operate 60 per cent of the time in order to obtain satisfactory operation. Determine the probability that (a) condensing unit will be on and operating all evaporators at same time, (b) that condensing unit will be on and operating one or more evaporators but not all three at same time, and (c) determine total running time of condensing unit.

SOLUTION:

(a) Probability of condensing unit on and operating all evaporators at same time =

$$\frac{I^n}{100^n} = \left(\frac{60}{100} \right)^3 = 0.22 \text{ or } 22\%$$

(b) Probability of condensing unit on and operating one or more, but not all, evaporators at same time =

$$1 - \left[\frac{I^n}{100^n} + \frac{(100 - I)^n}{100^n} \right] = 1 - \left[\left(\frac{60}{100} \right)^3 + \left(\frac{40}{100} \right)^3 \right] = 0.72 \text{ or } 72\%$$

² Elliott, James, "Multiple Systems or Individual Units." *Refrigerating Engineering*, Vol. 44, No. 4 (October, 1942), p. 237.

(c) Total running time =

$$100 \left[\frac{-(100 - I)^n + 100^n}{100^n} \right] = 100 \left[\frac{-40^3 + 100^3}{100^3} \right] = 94\%$$

In this example, if the three evaporators are multiplexed but thermostat-solenoid combinations are not used, the operating time of the condensing unit will be 60 per cent instead of 94 per cent.

It is evident that a series of evaporators may be operated from a single condensing unit with good temperature and load control by such a system, but at the expense of higher operating costs, longer running times, and more expensive controls. Power demands are reduced if individual condensing units are used for each evaporator but with the disadvantage of a somewhat higher initial installation cost.

14.11. Condensing-Water Control. The majority of refrigeration systems other than those of fractional horsepower use water-cooled rather than air-cooled condensers because the lower condensing temperatures result in more economical operation. Automatic control of the water flow to the condenser must be maintained if water wastage is to be eliminated. Such control may be provided through the use of either an *electric solenoid* water valve, similar in design to that described in §14.8, or by means of a *pressure-controlled* valve. With a solenoid valve the flow is two-position, either off or on, and its operation is simultaneous with starting and stopping of the compressor motor. With a pressure-operated water valve the flow is modulated and is dependent entirely upon condenser pressure rather than upon condensing-unit operation. For these reasons pressure-operated water valves, such as shown in Fig. 14.17, are becoming increasingly popular. Here two forces control the size of the valve opening through which condenser water may flow. An adjustable compression spring tends to keep the valve closed while a bellows, connected by tubing to the high pressure side of the system, tends to unseat the valve. When the condensing unit starts operation, the increased pressure in the condenser caused by noncondensed refrigerant gas opens the valve and permits water flow to the condenser. When condensing-unit operation has ceased, the water continues to flow through the valve until the condensing pressure has decreased below the valve setting. Similar water valves controlled thermostatically by the temperature of the water discharged from the condenser are available.

14.12. Compressor-Capacity Control. There are many refrigeration applications in which the refrigerating load is not constant, and therefore from the standpoint of both operation and economy, it is desirable to provide for some control of the capacity of the system. Several satisfactory means for obtaining such control are outlined in the following paragraphs.

Machine-speed adjustment is a comparatively simple yet satisfactory method of controlling capacity. Two-speed motors are satisfactory on

smaller installations and allow for fixed reductions of from one-third to one-half. Variable-pitch pulleys may be used for speed reduction down to one-half. Hydraulic couplings are obtainable that allow automatic control from full speed down to one-quarter speed. If the motor power is supplied through a reciprocating steam engine or a steam turbine, such units lend themselves to a reasonable degree of automatic speed control.

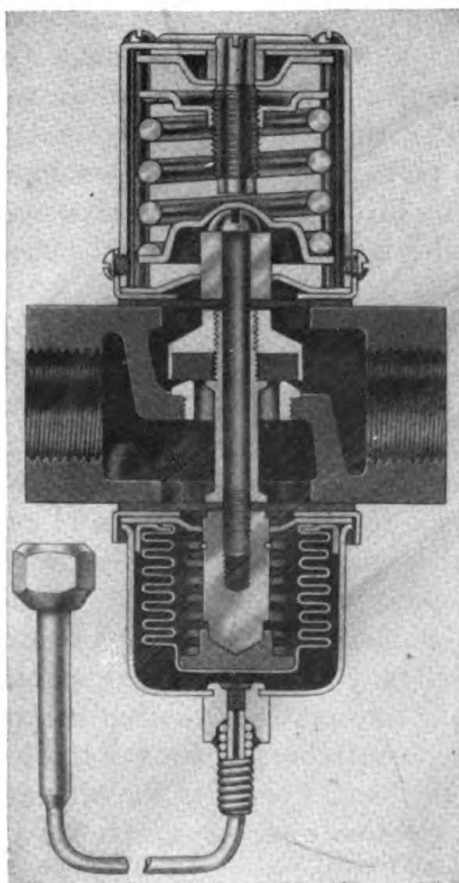


Fig. 14.17. Pressure-operated water valve. Courtesy Penn Electric Switch Co.

Clearance pockets are sometimes used to provide for a capacity reduction by increasing the clearance volume at the head end of the cylinders. Such reduction of compressor pumping capacity by the introduction of clearance spaces (discussed in Chapter 4) results from the increase in volume of space required by the clearance gases reexpanded to suction pressure. The introduction of additional clearance pockets or chambers may be accomplished manually or automatically.

Suction-line throttling is a simple and frequently used method of capacity control. This throttling is essentially what occurs when a back-pressure regulating valve is placed in the suction line of a single- or a

multiple-evaporator installation to provide for constant evaporator pressure and temperature. Although simple in application and operation, such a means of reducing compressor capacity is uneconomical in operation (see Chapter 13).

Some degree of capacity control may be obtained by making a portion of a cylinder ineffective or by making one or more cylinders of a multiple-cylinder compressor inoperative. This effect may be accomplished by means of a *cylinder by-pass* by providing a connection from the suction line to a port on the side wall of the cylinder. The cylinder will then become partially ineffective, since a portion of the gas is pushed back out through the side-wall suction port, and only that portion contained in the travel above this port is compressed. A complete cylinder may be rendered inoperative by lifting the suction-valve plate so that all the gas drawn into that cylinder is released back to the suction line with only a small loss of work.

Individual refrigeration machines may be used to provide capacity control by selecting their size so that the operation of all units will provide the maximum desired capacity, and operation of various combinations of the units will permit efficient capacity reduction. When this method of control is used, the units are usually installed with common suction and discharge headers. If, for example, a total capacity of 25 tons of refrigeration is required, control in 5-ton steps may be provided by the use of one 5-ton unit and two 10-ton units.

With practically all methods of capacity control described in this section, the control itself may be accomplished manually or automatically. One means of providing automatic regulation is through the use of electrical or pneumatic step controllers which will bring on additional capacity steps as progressive changes occur in the temperature of the load.

14.13. Miscellaneous Controls. A great many additional controls are constructed for special applications. Many of these are designed not to aid in the proper operation of the system but to prevent damage in case of improper operation. One such safety control is the *high-pressure cutoff*, frequently combined with a low-pressure motor control (see §14.7). Another safety control often used is a *low-voltage cutoff* that will shut down the system automatically when the line voltage drops below a minimum value. *High-pressure relief valves* are used for safety purposes to prevent damage in case excessive condensing pressures are encountered. *Oil separators* are often installed between the compressor and the condenser to prevent excessive oil pumping from the crankcase into the condenser and the evaporator. Separation of the oil from the refrigerant gas is usually accomplished by slowing down the gas velocity sufficiently to allow the oil to separate out by gravity. A float-operated valve maintains a set oil level in the separator, and the excess is forced from the

separator to the compressor crankcase by pressure difference. Figure 14.18 shows a typical oil separator.

In order that domestic refrigerators may be as completely automatic and free from attendance as possible, they require some form of automatic *defrosting* control. Some units incorporate a device that allows the refrigerator to operate at a temperature sufficiently high to permit defrosting yet low enough to prevent food spoilage during the defrosting cycle. In some units the starting of defrosting is manual, and the return to the normal operating cycle is automatic. Still a third device incorporated in some units is an automatic timing arrangement that puts the unit through a defrosting cycle every 24 hours.

Many special controls are necessary for the less frequently encountered systems of refrigeration such as air-cycle, steam-jet, centrifugal, and absorption. Small, continuously operated absorption systems such as domestic units require a thermostatic valve modulating the gas flow to the burner in order to control the amount of heat absorbed in the evaporator. Large absorption systems require, of course, more extensive controls. The principal control in many centrifugal refrigeration systems consists of a thermostatic element placed in the evaporating refrigerant and designed to modulate a valve in the water-supply line to the condenser. An increasing or decreasing load in the evaporator, as indicated by a rise or fall in the evaporator temperature, thus automatically changes the rate of refrigeration by modulating the supply of condensing water.

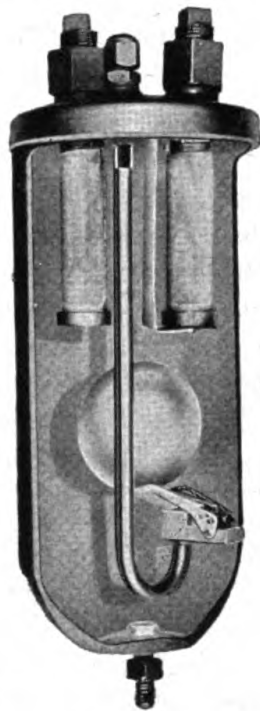


Fig. 14.18. Oil separator. Courtesy American Injector Co.

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PROBLEMS

14.1. A thermostatic expansion valve supplies Freon-12 to a coil evaporating refrigerant at 16 F. If the thermostatic bulb is also charged with Freon-12 and the valve is adjusted to open with 8 F superheat, determine (a) the force in the restraining valve spring and (b) the degrees superheat upon which the valve will open if the evaporating temperature is raised to 36 F.

14.2. If the thermostatic expansion valve of Problem 14.1 were placed in a system with methyl chloride evaporating at 16 F, what would be the condition of the refrigerant discharged from the evaporator coil?

14.3. An ammonia refrigerating system develops 10 tons when operating at 85 F condensing and 5 F evaporating with no liquid subcooling or vapor superheating. Determine (a) cubic feet per minute of refrigerant entering expansion valve and (b) cubic feet per minute leaving expansion valve.

14.4. If in Problem 14.3 the pressure drop through the liquid line between condenser and expansion valve is 3 psi, determine (a) quality of refrigerant entering expansion valve, (b) cubic feet per minute of refrigerant entering expansion valve, and (c) cubic feet per minute of refrigerant leaving expansion valve.

14.5. A condensing unit discharges saturated liquid refrigerant at 80 F from the condenser. If the evaporator and expansion valve are located 20 ft above the condenser, what is the quality of the refrigerant entering the expansion valve when the refrigerant is (a) Freon-22? (b) methyl chloride?

14.6. If the capacity of an expansion valve in tons of refrigeration is directly proportional to the orifice area and is based upon the volume of saturated liquid refrigerant, determine the valve capacity required to supply 10 tons under the conditions of Problem 14.5.

14.7. Five evaporators are operated with a single condensing unit and thermostat-solenoid and suction-line pressure-throttling controls. If three of the evaporators require a 60 per cent running time and two a 40 per cent running time, determine (a) percentage of condensing-unit running time with all evaporators in operation, (b) percentage of running time with one or more but not all evaporators in operation, and (c) total condensing-unit running time.

14.8. A simple field test for a thermostatic expansion valve consists in connecting the inlet to a supply of dry air (75 to 100 psi) or a service drum of Freon or methyl chloride. The thermostatic bulb is placed in a bath of melting ice, and a low-pressure gage is screwed loosely into the valve outlet to permit some leakage. If the valve is to be adjusted for 10 F superheat, determine the gage reading when adjusted for the refrigerant (a) Freon-12, (b) methyl chloride. (c) If the gage is now screwed tightly to stop all leakage, how should the pressure vary?

CHAPTER 15

Absorption Refrigeration and Chemical Dehumidification

15.1. History and Introduction. Since the most prevalent types of cooling or dehumidifying systems today employ a refrigerant compressor, the other methods used for refrigeration are not so well known. However, in many applications one of the less prominent systems may be the most feasible or practical, either economically or otherwise. Of these other methods used, the most common is the absorption refrigeration system, developed by Ferdinand Carré in France and on which he took out a patent in the United States in 1860. In the early years of the twentieth century absorption refrigeration gained considerable prominence, but after 1915 the electric-motor-driven, fully enclosed ammonia compressor was more actively promoted and received wide acceptance. Development work was then concentrated on compression systems, and absorption systems were practically forgotten, except for domestic units, until the late 1930's.

Since that time one company has introduced absorption air-conditioning units of 5, $7\frac{1}{2}$, 10, and 20 tons nominal capacity, and another has sold 3- and 5-ton units that have been further improved since World War II. Other companies have made some changes in the basic design of the old-style units and have installed several large industrial systems from 10 to 3600 tons capacity. One concern made and installed some very compact 150-ton units after World War II, primarily for comfort cooling.

The purpose of a refrigerant compressor is to withdraw the vapor from the evaporator, then to raise its temperature and pressure level so that it can reject in the condenser the heat absorbed in the evaporator and be ready to expand to the evaporator condition again. In an absorption system the refrigerant vapor is drawn from the evaporator by absorption into a liquid having a high affinity for the refrigerant. Applying heat to the solution raises its temperature level and drives off the refrigerant vapor so that it can pass on to the condenser to be liquefied. The liquid refrigerant then flows into the evaporator at a reduced pressure, and finally completes the cycle, being absorbed again by the absorbent, which flows through its own cycle.

The absorption of a gas or vapor by a liquid is theoretically governed by Raoult's law, which states that at a given temperature the ratio of the partial pressure of a volatile component in a solution to the vapor pressure of the pure component at the same temperature is equal to its mol

fraction in the solution. The mol fraction is equal to the number of mols of the component divided by the total number of mols present. The number of mols is equal to the given weight of a substance divided by its molecular weight.

Raoult's law applies only to an ideal solution or one in which the intermolecular forces between the substances in the solution are equal. Since there are no ideal solutions, deviations from Raoult's law exist and are termed positive or negative. A positive deviation exists when the observed pressure is greater than calculated, and a negative deviation when the observed pressure is less than calculated.

A desirable combination for an effective absorption refrigeration system would have a large negative deviation, which would require a minimum amount of absorbent for the given refrigerant quantity needed to circulate in the system. The less absorbent used, the smaller the amount of heat input required per ton; and a minimum heat input is desired.¹

The volume of a solution is not necessarily equal to the sum of the volumes of its components. In the case of ammonia and water, the total volume is approximately equal to the volume of water alone plus 85 per cent of the volume normally occupied by the ammonia.

The systems for large industrial or comfort cooling employ a pump to handle the absorbed refrigerant, but domestic refrigerators and the small units for air conditioning have no mechanical moving parts. Each of these systems will be described in the following sections. It is suggested that §2.8 be reviewed before proceeding further.

15.2. Simple Absorption System. The minimum number of primary units essential in an absorption system include an evaporator, absorber, generator, and condenser. An expansion valve, pressure-reducing valve, and a pump are used in a conventional two-fluid cycle, but the pump can be eliminated by adding a gaseous third fluid, as will be explained in §15.6. These units are connected to form the simple absorption cycle as diagramed in Fig. 15.1. Note that this cycle differs from a vapor compression cycle by the substitution of an absorber, generator, pump, and reducing valve for the compressor. Various combinations of fluids may be used, but that of ammonia and water is the most common and will be used in this discussion. The solution is called *aqua ammonia*; a strong solution contains about as much ammonia as possible; a weak solution contains considerably less ammonia.

The weak solution containing very little ammonia is sprayed or otherwise exposed in the absorber and absorbs ammonia coming from the evaporator. Absorption of the ammonia lowers the pressure in the absorber, which in turn draws more ammonia vapor from the evaporator. Usually

¹ *Heating Ventilating Air Conditioning Guide 1946*. New York: American Society of Heating and Ventilating Engineers, pp. 700-701.

some form of cooling is employed in the absorber to remove the heat of condensation and the heat of solution evolved there. The strong solution is then pumped into the generator, which is at a higher pressure and is where heat is applied. The heat vaporizes the ammonia, driving it out of solution and into the condenser, where it is liquefied. The ammonia passes on to the receiver, if a separate one is used, on through the expansion valve and into the evaporator. The weak solution left in the generator after the ammonia has been driven off flows through the reducing valve back to the absorber.

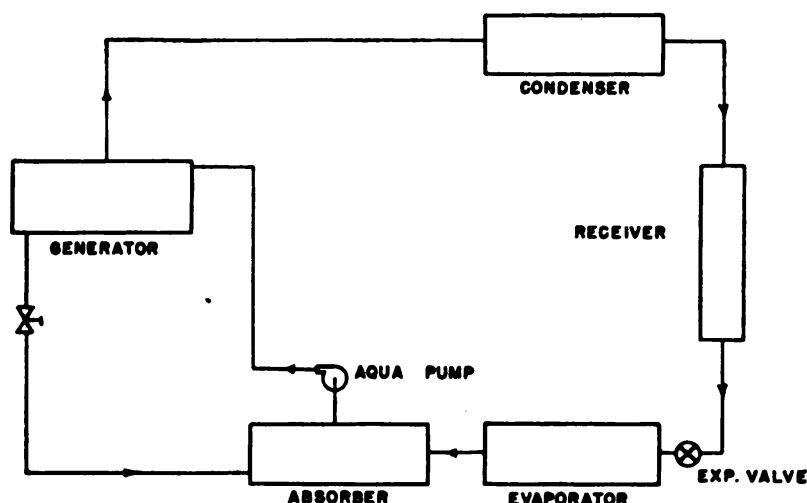


Fig. 15.1. Simple absorption system.

15.3. Practical Absorption System. The simple system described in the preceding section is not very economical. However, by the addition of certain auxiliary equipment the economics can be sufficiently improved to make the system practical. In order to obtain the most desirable balance between first cost and operating cost, the number of auxiliaries to use varies with the application. The basic auxiliaries include an analyzer, a rectifier, and two heat exchangers, connected as diagrammed in Fig. 15.2.

When ammonia is vaporized in the generator, some water is also vaporized and will flow into the condenser with the ammonia in the simple cycle, causing an *aqua solution* instead of *dry ammonia* to circulate through the expansion valve and evaporator. The analyzer and rectifier are added to separate the vapors leaving the generator and to send only dehydrated ammonia to the condenser. The aqua-ammonia heat exchanger, located between the absorber and generator, provides some cooling of the weak solution and heating of the strong solution. This operation saves on the amount of cooling needed for the absorber and the amount of heat needed for the generator, a very effective economy. The liquid sub-cooler, not always used, serves the same purpose as does a liquid-suction heat exchanger in a vapor compression system by subcooling the refrigerant.

erant entering the expansion valve. Sometimes the solution leaving the absorber is directed through a water-cooled heat exchanger. The basic units and important auxiliaries are described in the following paragraphs.

Absorber. In industrial systems the absorber may consist of one or more shells not unlike those of a shell-and-tube condenser. In some absorbers the weak solution from the generator and the vapor from the evaporator are introduced at the bottom and the strong solution is drawn off from the top. In others the weak solution is distributed across the top and flows down over the water-cooling tubes to the bottom, where as a strong solution it is drawn off. The ammonia vapor in this case is intro-

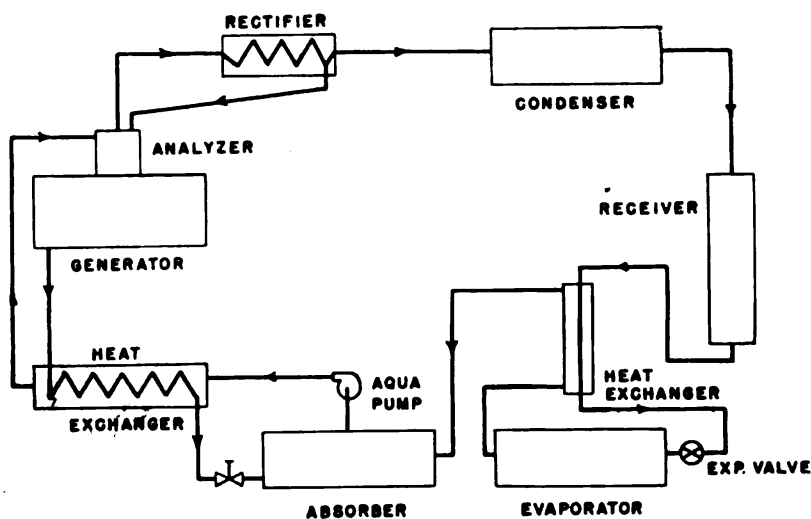


Fig. 15.2. Practical absorption system.

duced at about the middle of the horizontal shell. When multiple shells are used, they may be placed one above the other, and the solution flows down from one to the next with vapor being introduced into each shell. The cooling water may flow upward from shell to shell in series. Double-tube absorbers have also been used but are considered obsolete.

The purpose of the cooling water in the tubes is to remove the heat of condensation and heat of absorption as the vapor condenses and is absorbed by the weak solution. Keeping the aqua ammonia as cool as possible permits it to absorb more vapor and to become a stronger solution.

Generator. In the generator, energy is added to the system from an external source, usually being obtained from steam condensing in a coil or possibly from a direct-fired gas burner. The generator unit may consist of a horizontal shell partly filled with strong aqua automatically maintained at a constant level and covering the submerged steam or hot gas tubes. Connections must be provided for the strong aqua from the

absorber and the return solution from the rectifier to enter and for the vapor and the weak aqua to leave.

Another type consists of several vertical shell-and-tube heaters with aqua circulating through small tubes surrounded by steam introduced near the top. The tops of the heaters are connected to a common horizontal surge drum that has individual drop legs extending to the bottom of each heater. Vapor leaves from the surge drum, and circulation in the unit approximates that in a water-tube steam boiler.

Analyzer. The analyzer unit, sometimes referred to as an "open"-type cooler, may be built as an integral part of the generator or made as a separate piece of equipment. Both the strong aqua from the absorber and the aqua from the rectifier are introduced at the top and flow downward, usually over trays and into the generator. In this way considerable liquid surface area is exposed to the vapor rising from the generator. The vapor is cooled and most of the water vapor condenses, so that mainly ammonia vapor leaves the top of the analyzer. Because the aqua is heated by the vapor, less external heat is required in the generator.

Rectifier. The rectifier is a "closed" type vapor cooler. Its purpose is to cool further the vapor leaving the analyzer so that the remaining water vapor is condensed, leaving only dry or anhydrous ammonia vapor to flow on to the condenser. The rectifier is generally water-cooled and may be of the double-pipe, shell-and-coil, or shell-and-tube type. Temperatures of 100 to 120 F will usually dehydrate the ammonia sufficiently. Lower temperatures will cause too much ammonia to go into solution and leave with the liquid "drip" that returns to the analyzer and generator.

Aqua Heat Exchanger. The heat exchanger, located between the absorber and the generator, serves to cool the weak aqua by heating the strong aqua. The strong aqua has the greater heat capacity per pound and flows through in greater quantity than the weak aqua. Double-pipe exchangers are used in the small systems, whereas shell-and-tube construction is preferred for the large-capacity plants.

Analyses of some systems have indicated that removal of the water-cooled weak-aqua cooler and the installation of more surface in the aqua-to-aqua heat exchanger make for improved economy.

Detailed descriptions of many actual systems may be found in the literature.^{2,3} A typical flow diagram is shown in Fig. 15.3.

15.4. Refrigerant Absorbent Combinations. Properties of refrigerants were discussed in Chapter 5. Table 15.1 lists desirable properties of a solvent and of solvent-refrigerant combinations for absorption refrigeration. The items influenced by the characteristics of each property

² Taylor, R. S., "Heat Operated Absorption Units." *Refrigerating Engineering*, Vol. 49, No. 3 (March, 1945), p. 188.

³ Taylor, R. S., "Heat Operated Refrigerating Machines of the Absorption Type." *Refrigerating Engineering*, Vol. 17, No. 5 (May, 1929), p. 136.

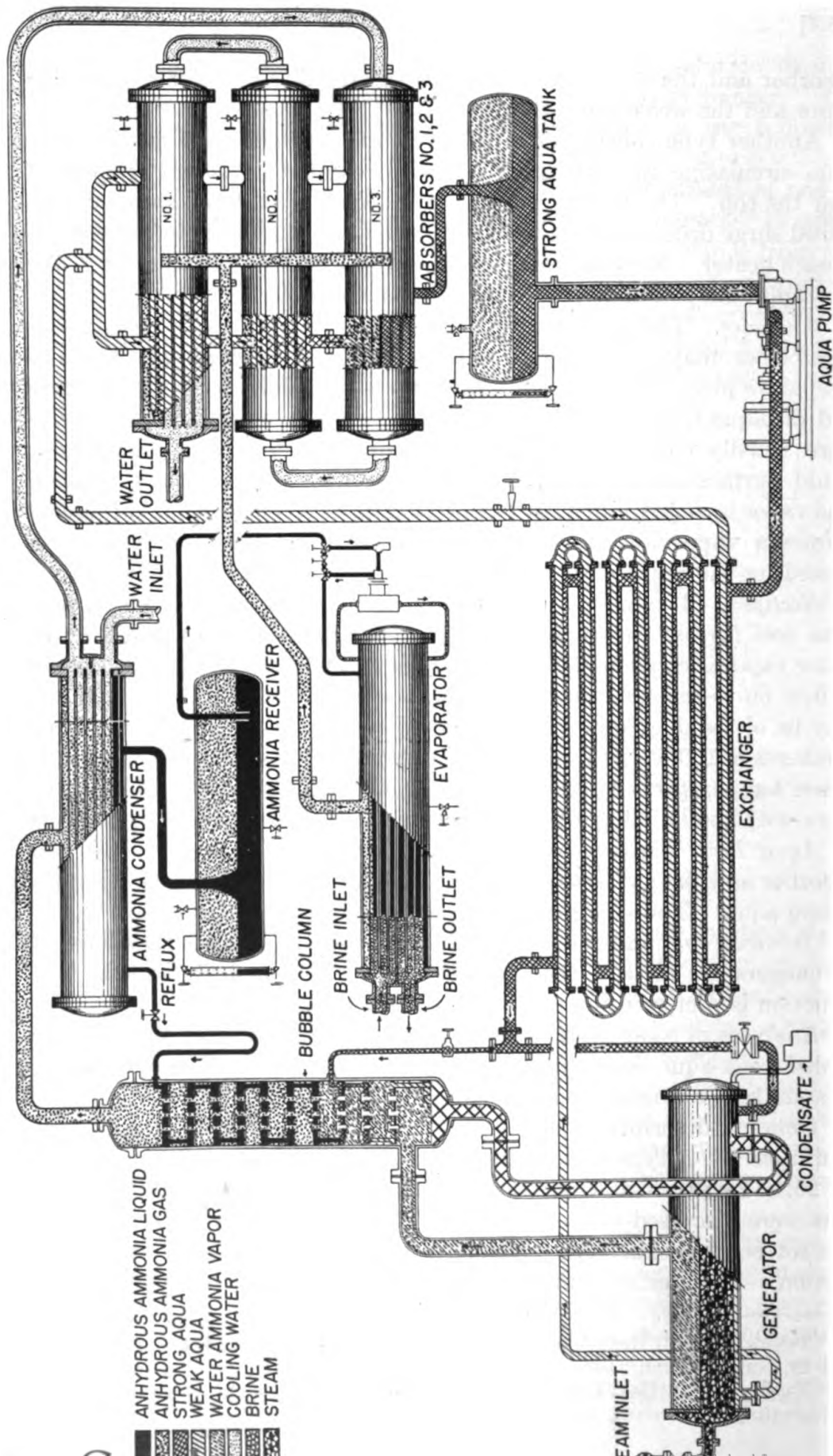


Fig. 15.3. Typical absorption-flow diagram. Courtesy York Corp., York, Penna.

are also indicated. These data, together with some of the following information were presented by Hainsworth.⁴

TABLE 15.1
THE ABSORPTION SYSTEM: DESIRABLE PROPERTIES AND CHARACTERISTICS⁵

<i>Solvent</i>	<i>Influence</i>
1. Vapor pressure at generator temperature should be negligible in comparison to the vapor pressure of the refrigerant at 100 F	Rectifier losses and operating cost
2. Temperature pressure concentration relations	Solvent should remain liquid throughout the cycle. Relations must be in confirmation with practical condenser, absorber and generator temperatures and pressures. Rate of circulation
3. Stability	Capable of withstanding heating operation at the maximum temperatures encountered in the generator
4. Low specific heat	Heat transfer requirements
5. Low surface tension	Heat transfer and absorption
6. Low viscosity	Heat transfer, power for pumping
<i>Solvent-Refrigerant Combination</i>	<i>Influence</i>
1. Solubility	High solubility of the refrigerant at temperatures of cooling medium (air or water) and at a pressure corresponding to the vapor pressure of the refrigerant at 40 F. Low solubility of the refrigerant in the solvent at generator temperatures and at a pressure corresponding to the vapor pressure of the refrigerant at temperature of cooling medium (air or water)
2. Stability	Solvent and refrigerant must be incapable of any non-reversible chemical action with each other within a practical temperature range (20 F to 250 F)
3. Superheating and supercooling	Affects operation

The absorbent should have a greater affinity for the refrigerant than the ordinary law of solubility would indicate. Very little heat is liberated when the Freons, nitrogen, or certain other gases are dissolved in water. However, water has a high affinity for ammonia, and considerable heat is evolved during the absorption. Thus the quantity of heat liberated in absorption is a crude measure of the affinity. The desirable absorbent refrigerant combination should have this property of high solubility at conditions in the absorber but should have low solubility at conditions in the generator.

As with refrigerants for compression systems, there is no single combination that excels in all the desirable properties. The choice depends

⁴ Hainsworth, W. R., "Refrigerants and Absorbents." *Refrigerating Engineering*, Vol. 28, Nos. 3 and 4 (August and September, 1944), pp. 97, 201.

⁵ From Hainsworth, W. R., "Refrigerants and Absorbents." *Refrigerating Engineering*, Vol. 48, No. 3 (September 1944), p. 201.

upon the application and economics involved. Hainsworth listed 49 refrigerants coupled with various absorbents to give 180 combinations as possibilities.

Ammonia and water were used by Carré in 1860, and this combination still is the most common mixture today. Thermodynamic properties of this mixture have been published recently which should provide data to calculate, more accurately than before, the theoretical heat transfer in various parts of the system.⁶ Other mixtures used to some extent include water and lithium bromide, Freon-12 and ethyl ether of diethylene glycol acetate, and methylene chloride and dimethoxytetraethylene glycol.

15.5. Absorption vs. Compression. The following summary of the advantages of absorption systems over compression systems gives some indication of the types of applications where they should be considered.

1. Since the only moving part of the entire system is the aqua pump, the operation is essentially quiet and subject to very little wear. The pump motor, engine, or turbine is quite small compared with the driving equipment required for a compression system of the same capacity.

2. Absorption systems are usually designed to use steam, either at high pressure or at low pressure. Waste or exhaust steam from other equipment may be used. There is no need for any electric power, although the pump is usually motor-driven. Steam that is bled from a turbine for heating in winter is needed less or not at all in summer, when more steam may be needed for refrigeration.

3. An absorption unit can operate at reduced evaporator pressure and temperature by increasing the steam pressure to the generator, with little decrease in capacity, whereas the capacity of a compression system drops rapidly with lowered evaporator pressure. Some applications require different evaporator temperatures at different times of the month or year. To cite specific examples, a single-stage compressor at 35 psi gage suction and 165 psi gage condensing will have its capacity reduced from 100 to 47 tons if the suction pressure is lowered to 10 psi. A typical absorption unit designed for 100 tons at 35 psi suction and 165 psi condensing when supplied with steam at 3.5 psi gage will produce 100 tons as well at 10 psi suction if the steam pressure is increased to 20.7 psi.⁷

4. At reduced loads the absorption unit is almost as efficient as at full capacity. Load variations are met by controlling the quantity of aqua circulated and the quantity of steam supplied to the generator.

5. Liquid refrigerant leaving the evaporator has no bad effect other than slightly unbalancing the system temporarily. However, so-called

⁶ Scatchard *et al.*, "Thermodynamic Properties of the Saturated Liquid and Vapor of Ammonia-Water Mixtures." *Refrigerating Engineering*, Vol. 53, No. 5 (May, 1947).

⁷ *York Ammonia Absorption Systems*. York, Pa.: York Corporation, 1940.

liquid slopover is detrimental to a compressor and requires preventive measures in compression systems. In applications where frequent load fluctuations occur, such as oil refineries, this feature is important.

6. Absorption units can be built in capacities well above 1000 tons each, which is the largest size to date for single compressor units. Absorption systems, except for the domestic units and for the latest comfort

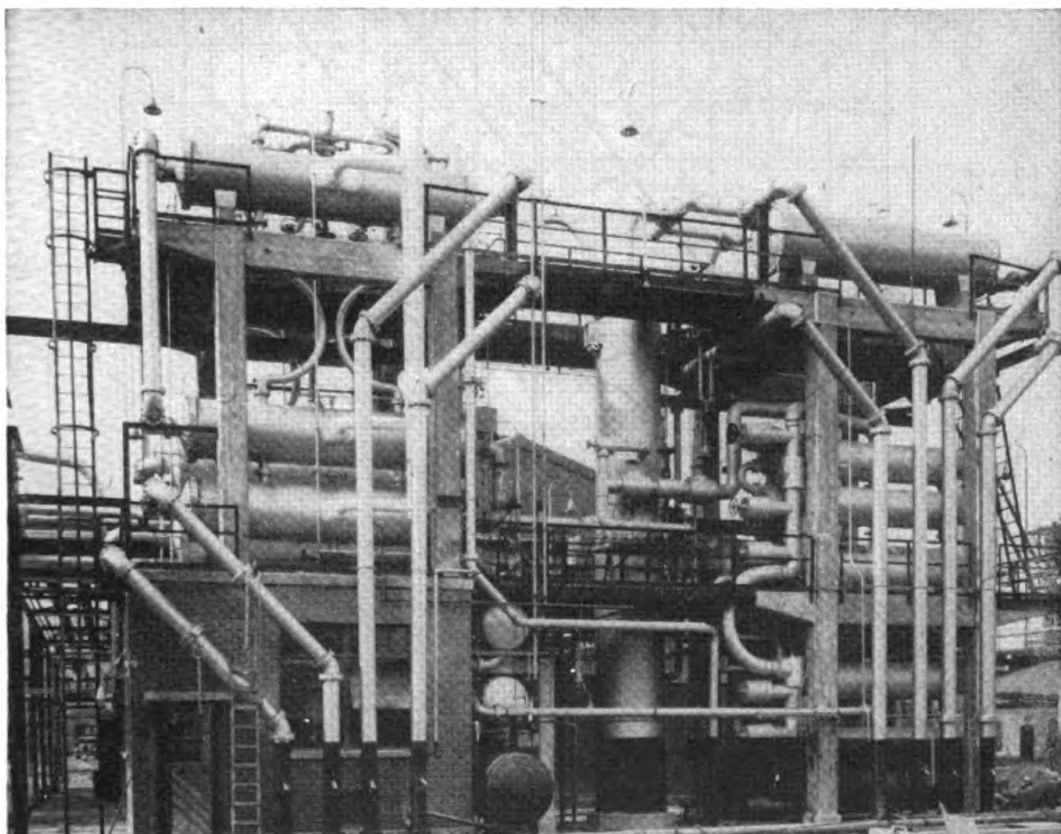


Fig. 15.4. 600-ton absorption unit. Courtesy York Corp., York, Penna.

cooling applications, generally take up more space than compression systems. However, the equipment can be located outdoors and assembled as a vertical unit requiring little ground area and no housing, as shown in Fig. 15.4.

7. Space requirements and automatic control requirements favor the absorption system more and more as the desired evaporator temperature drops.

Operating-cost comparisons for small units, presented by the American Gas Association,⁸ indicate that equal operating costs result for an electric compressor unit and a 3-ton absorption unit, each having a cooling tower,

⁸ "Gas Summer Air Conditioning." Cleveland: American Gas Association Testing Laboratories, Research Bulletin No. 18, 1943.

when electricity is 1.5 cents per kilowatt-hour and gas costs about 7 cents per 100,000 Btu, or with electricity at 2.5 cents and gas at about 11.5 cents. Lower gas rates or higher electric rates favor the absorption unit, whereas higher gas rates or lower electric rates favor the compression unit.

Without a cooling tower, equal operating costs result with electricity at 1.5 cents per kilowatt-hour and with gas at about 2 cents per 100,000

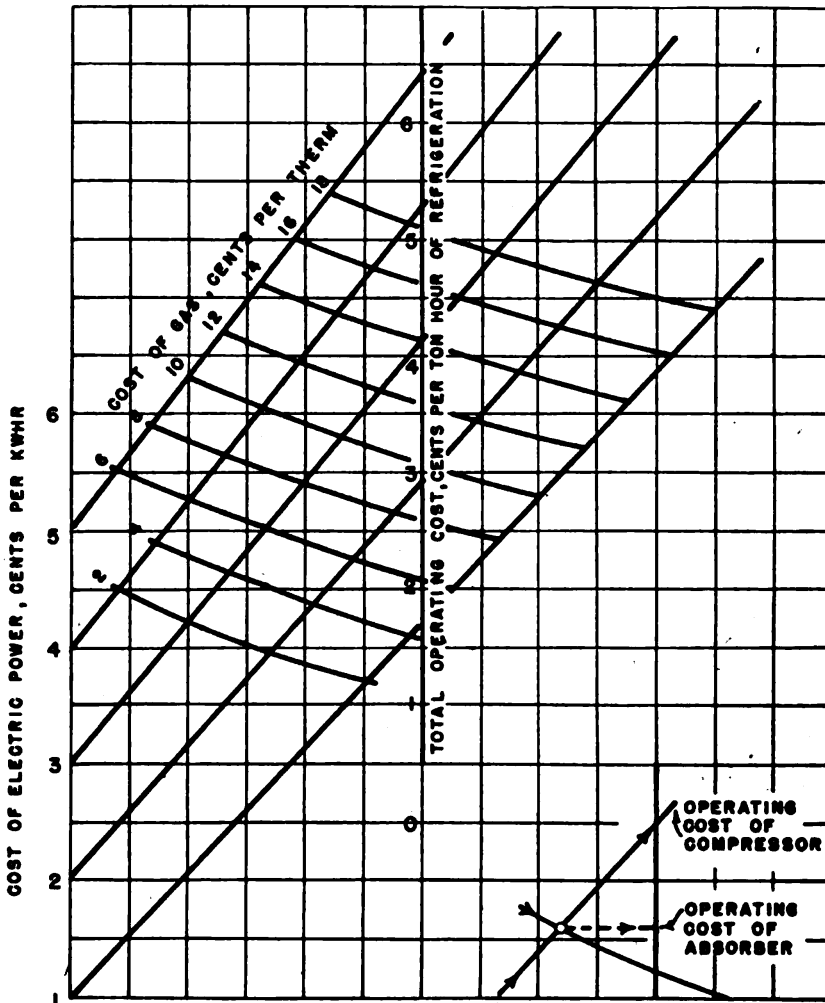


Fig. 15.5. Comparative operating costs of absorption- and electric-compression units with cooling tower. From AGA Research Bulletin No. 18, 1943.

Btu or with 2.5 cent electricity and about 8 cent gas rates. These comparisons and others are given in Figs. 15.5 and 15.6.

15.6. Domestic Unit. The domestic absorption-type refrigerator was developed from an invention by Carl Munters and Baltzer von Platen, who did their original work in 1925 while undergraduates at the Royal Institute of Technology in Stockholm. American rights were acquired from the Swedish Electrolux Company by Servel, Inc.

Ammonia is used as the refrigerant because it possesses most of the

desirable properties. It is toxic, but since the system has no valves or moving parts, there is very little chance for leaks, and the total amount of refrigerant used is small. Liquid ammonia evaporates in the presence of air or other gases; the lighter the gas, the faster the evaporation. Since hydrogen is the lightest gas and is also noncorrosive and insoluble in water, it is used in the *low side* of the system. Water is used as the solvent

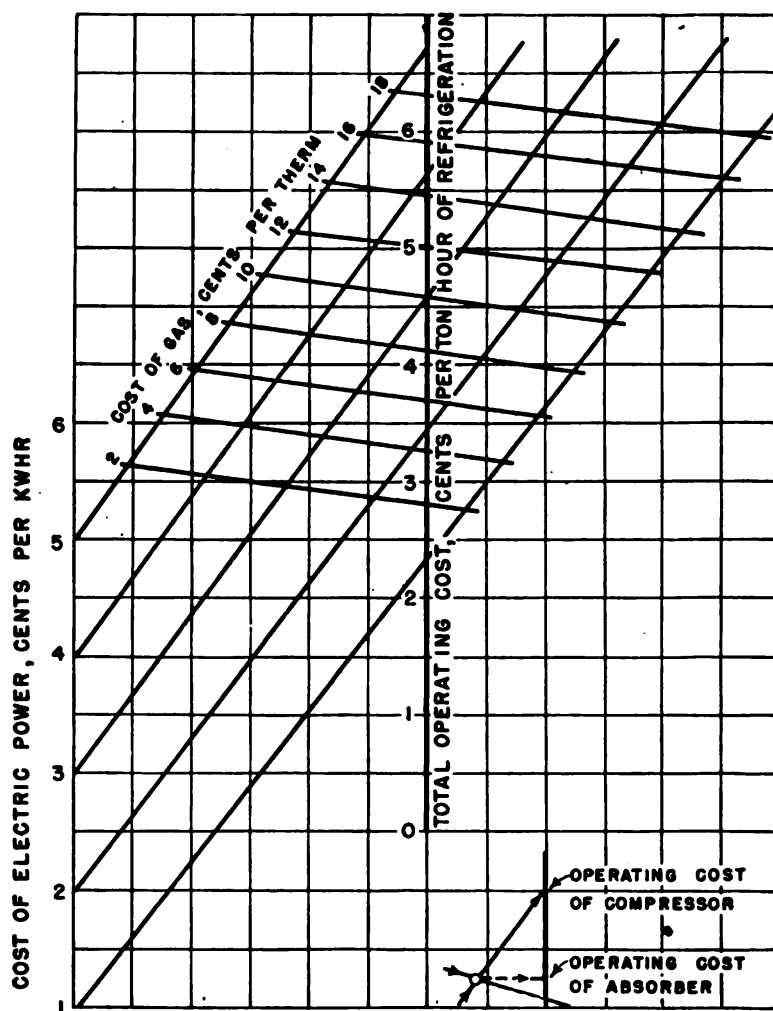


Fig. 15.6. Comparative operating costs of absorption- and electric-compression units without cooling tower. From AGA Research Bulletin No. 18, 1943.

because of its ability to absorb ammonia readily. Thus the three fluids used are ammonia, hydrogen, and water.

If liquid ammonia is introduced at the top of the system as shown in Fig. 15.7, it flows into the evaporator and evaporates. Hydrogen passes upward in the evaporator counterflow to the liquid ammonia that falls from one level to another. The ammonia vapor and hydrogen leave the top of the evaporator and pass through the gas heat exchanger, where they are warmed by the hydrogen flowing to the evaporator. They then

flow through the spherical vessel on the left and into the absorber. Water is introduced at the top of the absorber absorbing the ammonia as it passes counterflow through this unit. The hydrogen leaves the top of the absorber and flows through the heat exchanger on its way to the evaporator. The strong ammonia solution leaves the bottom of the absorber and flows into the generator at the lower right on the diagram.

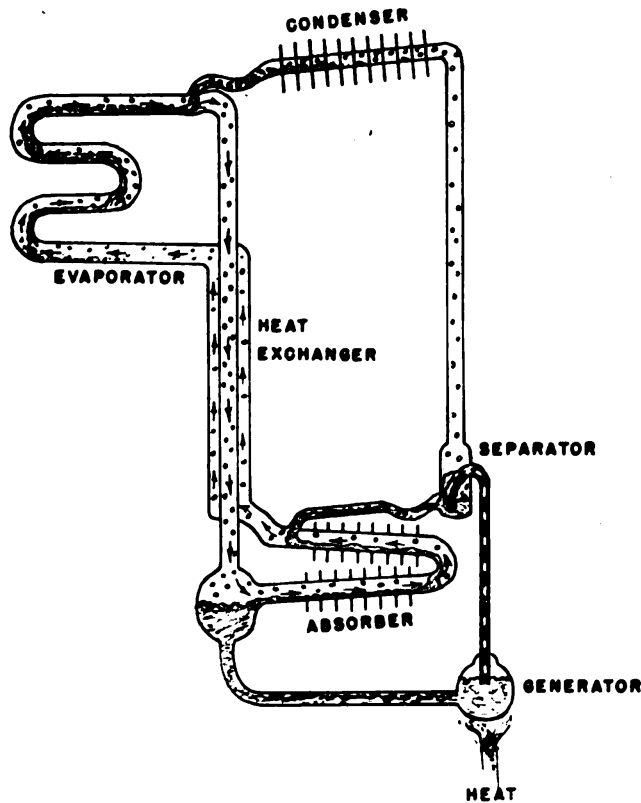


Fig. 15.7. Simple Electrolux system. (Courtesy Servel, Inc.)

Heat is applied at the generator, which drives the ammonia vapor out of solution. This vapor would easily rise into the condenser, but some means of elevating the remaining weak aqua so that it can flow into the top of the absorber must be used. The principle of the bubble pump is applied here: The discharge tube from the generator is extended down below the liquid level in the generator. As the vapor ammonia bubbles form and rise, they carry slugs of the weak ammonia solution with them up the discharge tube and into the separating vessel. From here the weak solution flows to the absorber to repeat its cycle, and the ammonia vapor flows to the gravity air-cooled condenser to be liquefied and then flows into the evaporator. Note the "U" bends in the weak solution line to the absorber and in the liquid line to the evaporator. These prevent the hydrogen from getting into the *high side* of the system.

The absolute pressure in the condenser is about the same as in the evaporator. Since practically pure ammonia is in the condenser, its vapor pressure there is substantially equal to the total pressure. In the

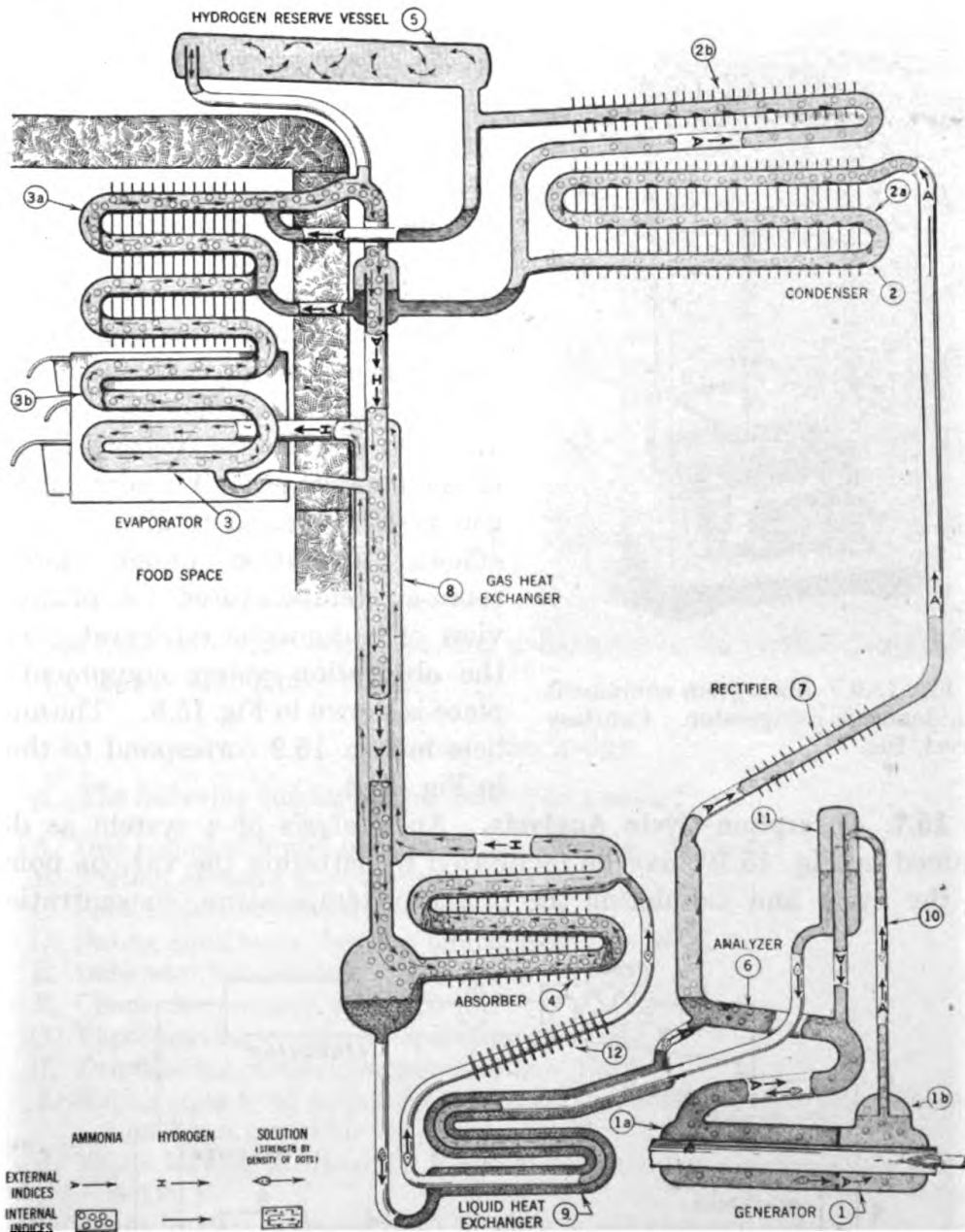


Fig. 15.8. Absorption refrigerator cycle. Courtesy Servel, Inc.

evaporator the ammonia-vapor pressure is much less and, in accordance with Dalton's law of partial pressures, is equal to the total pressure minus the partial pressure of the hydrogen. Being at a pressure below saturation pressure, the ammonia readily vaporizes in the evaporator and refrigerates.

The preceding paragraphs describe the basic units and operation of the Electrolux system. Actually, several refinements have been added to increase the efficiency and improve the performance, as shown in Fig.

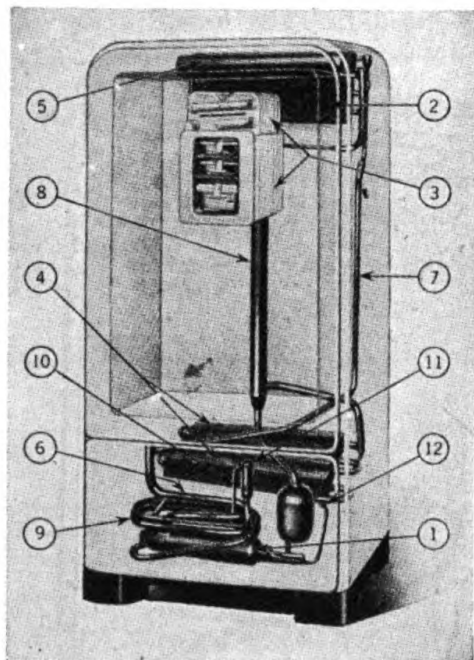


Fig. 15.9. Absorption equipment in a domestic refrigerator. Courtesy Servel, Inc.

15.8. A liquid heat exchanger is used for the weak solution going to the absorber and the strong solution going to the generator. The analyzer and rectifier are added to remove the water vapor that may have formed in the generator, so that only ammonia vapor goes to the condenser. The condenser and evaporator each consist of two sections, to permit extending the condenser below the top of the evaporator to provide more condenser surface, and to segregate the freezing portion of the evaporator. A reserve hydrogen vessel is added to give the same efficient operation under variable room-air temperatures. A phantom view of a domestic refrigerator with the absorption-system equipment in place is shown in Fig. 15.9. The numbers in Fig. 15.9 correspond to those in Fig. 15.8.

15.7. Absorption-Cycle Analysis. An analysis of a system as diagrammed in Fig. 15.10 may be facilitated by lettering the various points in the cycle and tabulating the pressure, temperature, concentration,

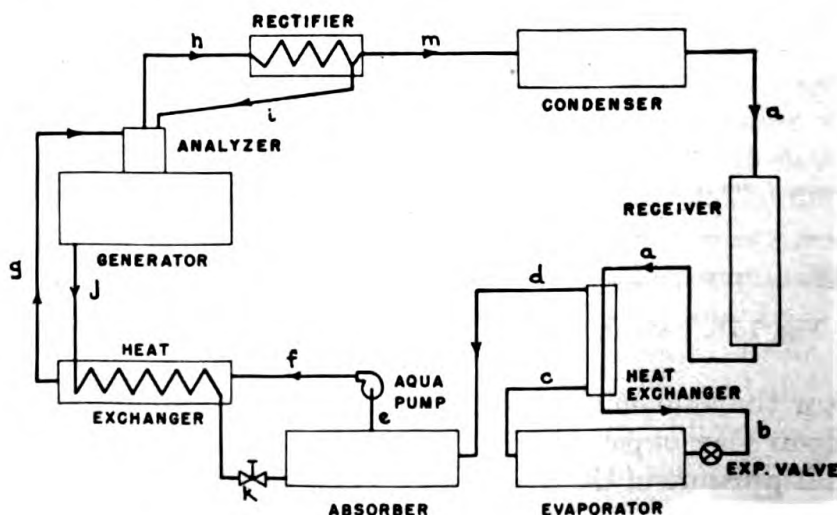


Fig. 15.10. Absorption refrigeration cycle.

enthalpy, and weight at each point. This procedure has been followed in Table 15.2, with the points of the system designated in the first column on the left. The numbers in the table refer to the following steps, in which the method of determining each value is explained. The general

TABLE 15.2

Point	Pressure, p	Temperature, t	Flow Weight, W	Concentration, x	Specific Enthalpy, h
<i>a</i>	1	8	1	6	8
<i>b</i>	3	..	4	6	20
<i>c</i>	1	1	4	6	18
<i>d</i>	3	9	4	6	17
<i>e</i>	3	1	12	6	7
<i>f</i>	3	..	12	6	16
<i>g</i>	3	1	12	5	5
<i>h</i>	3	1	14	5	5
<i>i</i>	3	..	13	5	5
<i>j</i>	3	1	11	5	5
<i>k</i>	3	..	10	6	19
<i>m</i>	3	1	4	5	5

method used here has been presented previously in the ASRE *Data Book*⁹ and by Raber and Hutchinson.¹⁰

ANALYSIS PROCEDURE

1. The following conditions are desired or assumed:
 - A. One pound of liquid per time unit leaves the condenser.
 - B. Suction pressure $p_c = 20$ psi.
 - C. Suction temperature $t_c = 10$ F.
 - D. Strong aqua from absorber temperature $t_a = 90$ F.
 - E. Generator temperature $t_g = 200$ F.
 - F. Condenser pressure $p_a = 155$ psi.
 - G. Vapor leaving rectifier temperature $t_m = 130$ F.
 - H. Drip leaving rectifier temperature $t_i = 130$ F.
 - I. Strong aqua from exchanger saturated at t_o which is 20 F less than weak aqua from generator at t_i , $t_o = t_i - 20 = 180$ F.
 - J. Vapor leaving analyzer 10 F above aqua entering analyzer $t_k = t_o + 10 = 190$ F.
 - K. Vapor leaving precoolers is 10 F below liquid from condenser.
 - L. Equilibrium concentration exists for vapor m leaving rectifier, vapor h leaving analyzer, aqua j leaving generator, aqua g leaving exchanger, and aqua a leaving condenser.
 - M. Pressure drop through each unit is 2 psi.

⁹ *Refrigerating Data Book*, 5th ed. New York: American Society of Refrigerating Engineers, 1943, p. 43.

¹⁰ Raber, B. F., and Hutchinson, F. W., *Refrigeration and Air Conditioning Engineering*. New York: John Wiley & Sons, Inc., 1945, Chapter V.

2. The necessary values for the analysis are tabulated as determined and shown completed in Table 15.3.

3. From steps 1B, 1F, and 1L the pressure at the various points may easily be calculated.

4. The rate of flow or weight at m , h , c , and d is the same as at a , which is 1 lb per unit time.

5. By reference to a chart of aqua-ammonia properties, such as the one in the Appendix, the concentration and enthalpy at g , h , i , j , and m may be read since the pressure and temperature at each point are known. Note that there is vapor at h and m and liquid at g , i , and j . $x_h = 0.962$, $x_i = 0.60$.

TABLE 15.3

Point	Pressure, p	Temperature, t	Flow Weight, W	Concen- tration, x	Specific Enthalpy, h	Total Enthalpy, H
a	155 psi	87 F	1 pound	0.994	137 Btu/lb	137 Btu
b	153	1	0.994	61	61
c	20	10	1	0.994	584	584
d	18	77	1	0.994	660	660
e	16	90	9.67	0.437	-15	-145
f	163	9.67	0.437	-7.8	74
g	161	180	9.67	0.437	87	841
h	159	190	1.097	0.962	718	788
i	157	130	0.097	0.600	54	5
j	155	200	8.67	0.372	108	936
k	153	8.67	0.372	2.2	19
m	157	130	1	0.994	667	667

6. Then concentration $x_a = x_f = x_g = 0.437$. $x_a = x_b = x_c = x_d = x_m = 0.994$ and $x_k = x_j = 0.372$.

7. The enthalpy of the subcooled liquid at e is read as heat of liquid opposite the concentration and temperature point. $h_e = -15$.

8. Enthalpy and temperature at a may be read from the liquid concentration and pressure values. $h_a = 137$, $t_a = 87$ F.

9. From 1K, $t_d = t_a - 10 = 77$ F.

10. Analyzing the absorber,

$$W_e = W_k + W_d \quad \text{and} \quad x_e W_e = x_k W_k + x_d W_d$$

By substitution,

$$x_e W_k + x_e W_d = x_k W_k + x_d W_d$$

or
$$W_k = \frac{W_d(x_d - x_e)}{(x_e - x_k)} = \frac{1(0.994 - 0.437)}{(0.437 - 0.372)} = 8.67 \text{ lb}$$

11. $W_j = W_k = 8.67 \text{ lb}$

12. $W_e = W_k + W_d = 8.67 + 1 = 9.67 = W_f = W_g$

13. Analyzing the rectifier,

$$W_h = W_i + W_m \quad \text{and} \quad x_h W_h = x_i W_i + x_m W_m$$

By substitution,

$$x_h W_i + x_h W_m = x_i W_i + x_m W_m$$

$$\text{or} \quad W_i = \frac{W_m(x_m - x_h)}{(x_h - x_i)} = \frac{1(0.994 - 0.962)}{(0.962 - 0.600)} = 0.097 \text{ lb}$$

$$14. \quad W_h = W_i + W_m = 0.097 + 1.0 = 1.097 \text{ lb}$$

$$15. \quad \text{The pump work} = \frac{144(p_f - p_c)W_o}{778 \times \frac{1}{v_o} \times \eta}$$

$$v_o = (1 - x_o) \text{ volume of water} + (0.85x_o) \text{ volume of NH}_3$$

Using steam and ammonia tables for the volumes,

$$v_o = (1 - 0.437)0.0161 + 0.85 \times 0.437 \times 0.02707 = 0.0191 \text{ cu ft}$$

Assuming pump efficiency as 70%,

$$\text{Pump work} = \frac{144(163 - 16)9.67}{778 \times \frac{1}{0.0191} \times 0.70} = 7.2 \text{ Btu}$$

$$16. \quad h_f = h_o + \text{pump work} = -15 + 7.2 = -7.8 \text{ Btu}$$

17. The total weight of ammonia at d is equal to the sum of the weights of ammonia in the aqua and in the vapor, or

$$x_d W_d - x_{da} W_{da} + (W_d - W_{da})x_{dv}$$

Knowing the pressure p_d and temperature t_d , the concentration and enthalpy of the aqua, x_{da} and h_{da} , and of the vapor, x_{dv} and h_{dv} , may be found from the chart.

From the above equation,

$$W_{da} = W_d \frac{(x_{dv} - x_d)}{(x_{dv} - x_{da})} = 1 \frac{(0.994 - 0.994)}{(0.994 - 0.427)}$$

and

$$h_d = W_{da}h_{da} + (W_d - W_{da})h_{dv}$$

But when x_d is equal to x_{dv} , as in this example, or greater than x_{dv} , the mixture at d is dry-saturated or superheated. Raber and Hutchinson have suggested the following empirical equation for such cases:

$$pd'' = \frac{17p(17 - 0.8x_o)(1 - x_o)}{(17 + x_o)}$$

where pd'' = partial pressure of water vapor over an aqua ammonia solution in equilibrium with vapor at x_d and p_d and is approximately equal to the partial pressure of the water in the superheated mixture at d .

p = saturation pressure of pure water at t_o

x_o = ammonia concentration in a solution in equilibrium with vapor at x_d and p_d

The enthalpy of superheated steam, h_d'' , at p_d and t_d may be calculated in accordance with the explanation given for equations 9.10 and 9.11.

The partial pressure of ammonia in the vapor is $p_d' = p_d - p_d''$.

The enthalpy of superheated ammonia, h_d' , is read from an ammonia table or chart at p_d' and t_d' .

The enthalpy of the mixture at d is therefore $h_d = x_d h_d' + (1 - x_d) h_d''$.

Substitution of values from our example gives

$$p_d'' = \frac{17 \times 0.46(17 - 0.8 \times 0.43)(1 - 0.43)}{(17 + 0.43)} = 0.248 \text{ psi}$$

$$h_d'' \text{ at } 0.248 \text{ psi and } 77 \text{ F} = 0.1061 + 0.444 \times 77 = 1095.1 \text{ Btu}$$

$$p_d' = 18 - 0.248 = 17.75 \text{ psi}$$

$$h_d' \text{ at } 17.75 \text{ psi and } 77 \text{ F} = 657 \text{ Btu}$$

$$h_d = 0.994 \times 657 + 0.006 \times 1095.1 = 660 \text{ Btu}$$

18. At c the entrained aqua quantity is

$$W_{ca} = W_c \frac{(x_{cv} - x_c)}{(x_{cv} - x_{ca})}$$

and

$$h_c = W_{ca} h_{ca} + (W_c - W_{ca}) h_{cv}$$

The concentration and enthalpy values may be read from the chart at p_c , t_c .

$$\text{Then } W_{ca} = 1 \frac{(0.9995 - 0.994)}{(0.9995 - 0.89)} = \frac{0.0055}{0.1095} = 0.052 \text{ lb}$$

$$\text{and } h_c = 0.052 \times 10 + (1 - 0.052) 615 = 584 \text{ Btu}$$

19. Relative to the heat exchanger,

$$W_k(h_i - h_k) = W_f(h_g - h_f)$$

or

$$h_k = h_i - \frac{W_f}{W_k} (h_g - h_f)$$

$$\text{Then } h_k = 108 - \frac{9.67}{8.67} (87 + 7.8) = 2.2 \text{ Btu}$$

20. Relative to the subcooler,

$$h_b = h_a - (h_d - h_c)$$

or

$$h_b = 137 - (660 - 584) = 61 \text{ Btu}$$

A heat balance can now be calculated for the system:

$$\text{Refrigerating effect } H_c - H_b = 584 - 61 = 523$$

$$\text{Heat added in generator } H_h + H_j - H_i - H_g = 788 + 936 - 5 - 841 = 878$$

$$\text{Energy added by pump} = 7$$

$$\text{Total heat added} = 1408 \text{ Btu}$$

$$\text{Heat rejected in absorber } H_d + H_k - H_e = 660 + 19 + 145 = 824$$

$$\text{Heat rejected in condenser } H_m - H_a = 667 - 137 = 530$$

$$\text{Heat rejected in rectifier } H_h - H_m - H_i = 788 - 667 - 5 = 116$$

$$\text{Total heat rejected} = 1470 \text{ Btu}$$

Unaccountable heat $1470 - 1408 = 62$ Btu

$$\frac{62}{1408} = 4.4\%$$

15.8. Other Systems. In addition to the absorption cycles described in §§15.3 and 15.6, there are multistage systems. One two-stage cycle uses two evaporators, each operating at a different pressure with two absorbers but with a single generator and one condenser. Another type

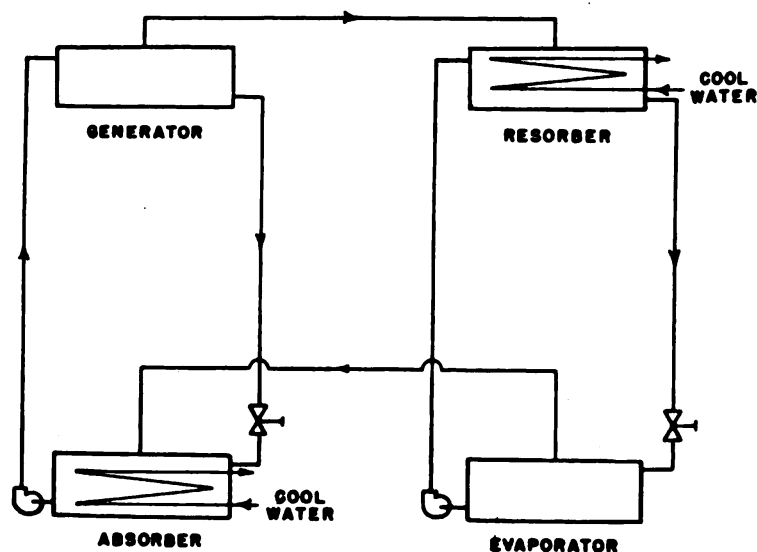


Fig. 15.11. Simple resorption system.

has two absorbers and two generators with one condenser and one evaporator. Still other combinations are possible.

There also is a *resorption system* as diagrammed in Fig. 15.11. The *condenser* is replaced by a *resorber*, in which the vapor is not condensed to a pure liquid but is absorbed by a special weak solution while condensing. This weak solution is circulated from the evaporator by a pump. As the vapor is condensed and absorbed in the resorber, heat is given up and is removed by the cooling water. The solution then flows through the expansion valve to the evaporator. As the ammonia boils in the evaporator, the heat of vaporization and the heat of disassociation both produce the refrigerating effect. More cooling effect can thus be produced than in the usual absorption system, but an extra pump is required. Combination resorption and adsorption cycles may also be used for special applications.

15.9. Chemical Dehumidification. Chemicals are sometimes used separately or in conjunction with refrigeration to remove undesired moisture from air, particularly for many industrial air-conditioning applications requiring either a low relative humidity or low dew-point temperature in the room. Straight refrigeration for many of these applications would not be economical or practical. The refrigeration engineer

should be familiar with these other methods used for dehumidification and should understand the principles involved, as well as the advantages and disadvantages. The two types of substances used for dehumidification are absorbents and adsorbents.

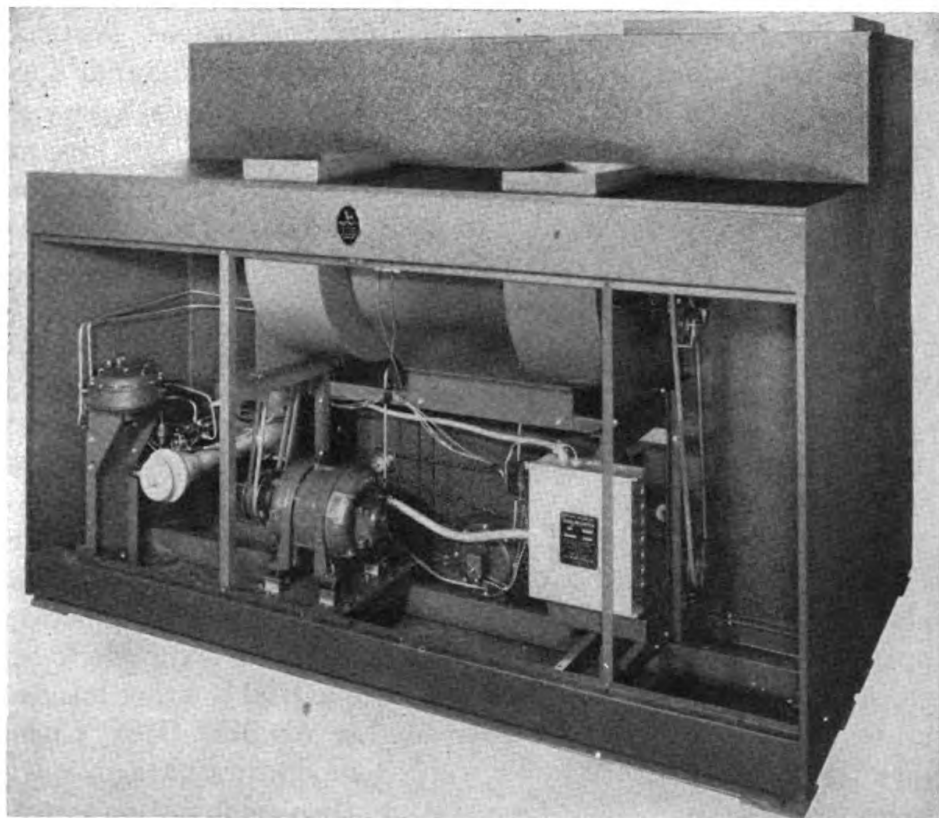
Absorbents are substances which can take up water vapor but which in so doing change chemically or physically, or in both respects. Materials commonly used include solid calcium chloride, which is employed principally in small cartridge-type driers or in desiccating chambers. Only the exposed surface is effective; it soon becomes pasty and then crusty, thus preventing air from contacting the material underneath. Water solutions or brines of calcium chloride, lithium chloride, lithium bromide, and ethylene glycol are used as air dehydrators by spraying or otherwise exposing a large surface of the solution in the air stream.

A desirable absorbent is capable of having its reduced vapor pressure controlled (obtained by controlling the concentration and temperature of the solution), is available at reasonable cost, noncorrosive, odorless, nontoxic, noninflammable, chemically inert against air impurities and equipment material, stable through range of use, low in viscosity, and capable of easy regeneration or reconcentration after absorbing the moisture. As moisture is absorbed, its latent heat is given up as well as the heat of solution, which heats the absorption solution and may also raise the temperature of the dehydrated fluid. In some air-conditioning applications it is necessary to pass the air over an aftercooler.

Adsorbents are substances in the solid state that contain numerous, minute pores into which moisture or other vapors from the surrounding medium are easily drawn or attracted. This process takes place without permanently changing the substance either chemically or physically. The vapors adsorbed can be driven off later by the application of heat, thus reactivating the substance so that it can be reused. The materials most commonly used are silica gel, which is a form of silicon dioxide prepared by mixing sulfuric acid with sodium silicate, and activated alumina, which is a porous amorphous form of aluminum oxide. A silica gel unit with a rotating gel bed is shown in Fig. 15.12. The left fan exhausts the gas-heated reactivation air leaving part of the gel bed. The right fan discharges the dehumidified air leaving the reactivated gel in the other portion of the rotating bed.

Desirable properties for an adsorbent include suitable vapor-pressure characteristics, availability at reasonable cost, absorptivity of sufficient moisture per pound of substance, chemical stability, physical ruggedness, and capability of repeated reactivation at reasonable temperatures.

15.10. Applications. An economic study often indicates that some means of dehumidification other than refrigeration would be more desirable in air conditioning when the load has a low sensible-heat factor or calls for a low apparatus dew point. Although absorption or adsorption



(a)

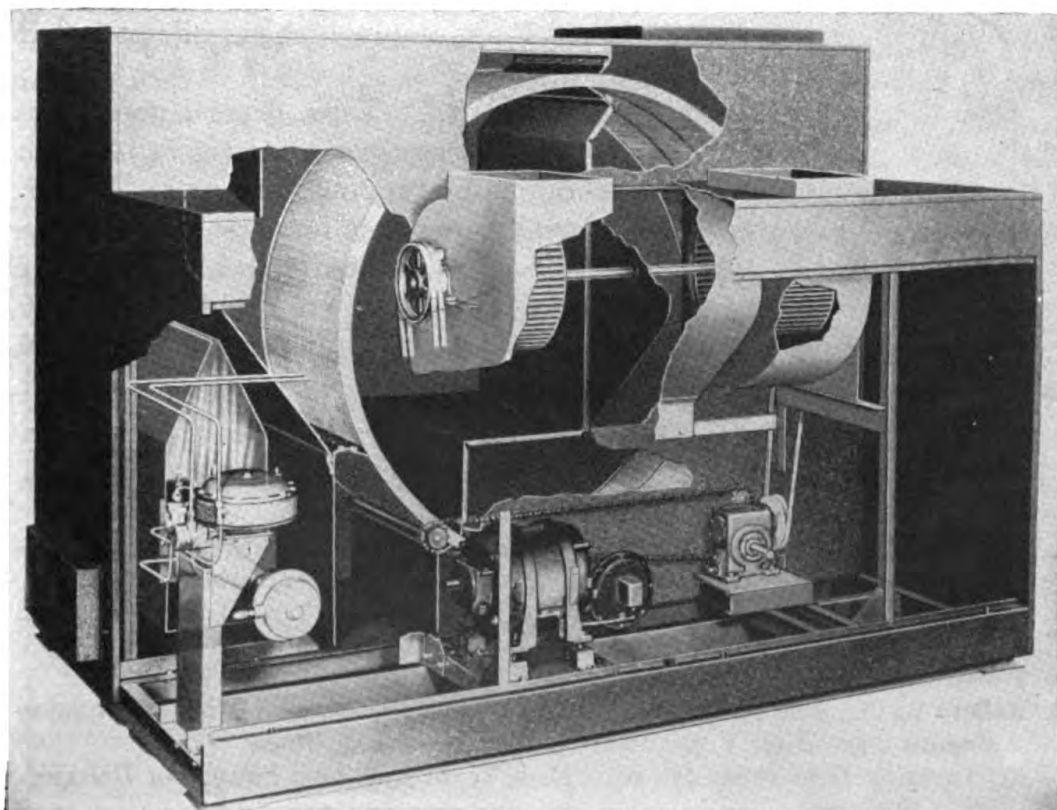


Fig. 15.12. Silica-gel adsorption unit. (a) With cover removed. (b) Cutaway view.

equipment has been used for a few comfort applications, particularly restaurants, the majority of the installations are for industrial applications. These include the manufacture of rugs, chemicals, safety glass, electrical equipment, plastics, and paper products; the drying of oils, leather, lacquers, and films; and air conditioning museums, libraries, vaults, and telephone exchanges. In each the control of humidity is of prime importance.

No set rule can be applied to determine whether refrigeration or some other form of dehumidification will be more economical. In general, it has been found that chemical dehydration should be used when a dew point under 32 F is required in a room with a dry bulb over 32 F, and that adsorption systems become favorable when the latent load is 30 per cent or more of the total load.

The following items should be considered, since they influence the first cost of dehumidification equipment: apparatus dew point, water availability and temperature, load variations, and room temperatures. An apparatus dew point above 40 F is generally more favorable to refrigeration, whereas one below 40 F may be more partial to other means. A water temperature of about 65 F is another dividing line. Chemical equipment is favored with a lower water temperature until it is within 5 F of the apparatus dew-point temperature; then combination water and refrigeration or all water cooling may be better. When constant humidity must be maintained under wide load fluctuations or when dry-bulb temperatures over 90 F or under 50 F are required, chemical equipment may be more favorable.

Operating costs must also be considered. These depend upon electricity and gas or steam rates. With water-cooled refrigeration, the water quantity used for each general method is about the same. Equipment using a liquid absorber is generally greater in first cost than adsorption or refrigeration equipment. It is best suited where low relative humidities and the high dry-bulb temperatures are desired, in which case the operating costs are generally less than for the other types of systems.

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PROBLEMS

15.1. Calculate the heat balance for an aqua-ammonia absorption system in which the suction pressure is 40 psi, suction temperature is 30 F, strong aqua from absorber is at 100 F, generator is at 210 F, condenser is at 165 psi, vapor and drip from rectifier are at 135 F, strong aqua from exchanger is at 190 F, and other general conditions are the same as in the example in §15.7.

15.2. Calculate the heat balance for an ammonia absorption system for the same conditions as given in Example 15.1 but with an evaporator suction temperature of 10 F and a pressure of 20 psi.

15.3. Compare the operating costs of a 3-ton electric compressor unit with that for a 3-ton absorption unit, each using a cooling tower, if electricity costs 3 cents per kilowatt-hour and gas of 500 Btu per cubic foot costs 50 cents per 1000 cu ft.

15.4. Compare the operating costs of 3-ton electric compressor and absorption systems without cooling towers if electricity costs 2.3 cents per kilowatt-hour and gas of 1000 Btu per cubic foot costs 70 cents per 1000 cu ft.

15.5. Determine whether chemical dehydration or refrigeration equipment would tend to be more favorable under the following conditions: (a) room conditions of 70 F dry bulb and 30 per cent relative humidity, available water temperature at 65 F; (b) room conditions of 60 F dry bulb and 40 per cent relative humidity with 65 F water.

CHAPTER 16

Low-Temperature Refrigeration

16.1. Introduction. The first low-temperature refrigeration systems were developed primarily for the solidification of carbon dioxide and the liquefaction and subsequent fractional distillation of such gases as air, oxygen, nitrogen, hydrogen, and helium. Oxygen was liquefied as early as 1877 by Coilletet and Pictet; Dewar, applying Joule-Thomson expansion of the gas, succeeded in liquefying hydrogen in 1898. At atmospheric pressure liquid oxygen boils at 90.2 K (-297.3°F) and liquid hydrogen at 20.4 K (-423°F). Liquefaction of helium was accomplished by H. Kamerlingh Onnes in 1908 in the famous cryogenic¹ laboratories of the University of Leiden. By evaporation of liquid helium under high vacuum, temperatures as low as 1.1 K (-457.7°F) were initially attained, and by improvements in the apparatus 0.7 K (-458.4°F) was reached by the year 1928. It was not until 1933 that Giauque and Debye independently proposed the adiabatic demagnetization of paramagnetic salts as a means of attaining lower temperatures, and through their methods 0.1 K was reached. To date the lowest temperature successfully reached by this method is 0.004 K.

By the use of modern measuring instruments, temperatures on the earth's surface ranging from 136 F (at Tripoli) to -90.4°F (at Verkhoyansk, Siberia) have been recorded; stellar temperatures ranging from 50,000,000 F at the center of the sun to 3 K in interstellar space have been approximately determined. As noted in the preceding paragraph, lower temperatures have been attained in the laboratory. Before studying the methods by which such temperatures can be produced, it is first desirable to review the evolution of the temperature scales together with modern concepts of heat. Without knowledge of the meaning of absolute zero a study of its experimental approach means little.

16.2. Temperature Scales. The struggle to attain increasingly higher and lower controlled temperatures in the laboratory has been a continuing one since the first modern concepts of heat were expounded. The American physicist Benjamin Thompson, Count Rumford, conducted his classical experiment of quantitative heat measurement while boring a cannon for the King of Bavaria in 1790; Helmholtz published the first

¹ The term "cryogenic" is derived from the Greek word *kryos*, meaning cold or frost, and is frequently applied to very low-temperature refrigeration applications like those encountered in the liquefaction of gases and in the study of physical phenomena at temperatures approaching absolute zero.

generalized concept of heat in his paper on the conservation of energy in 1847. Prior to that time the theory most adhered to was proposed by a Prussian physician, Stahl, who died in 1734. He presumed that heat was a material substance to which he gave the name of phlogiston.

Our most fundamental concepts of heat are obtained through the sensations experienced when we touch objects that we say are "hot" or "cold." Qualitative measurements of these sensations must be made, however, through the physical changes which occur in bodies when exposed to these different environments of temperature. Those most commonly applied are changes in volume (such as in a mercury thermometer), changes in pressure (as with the constant-volume gas thermometer), changes in electrical resistance (as with the resistance thermometer), changes in radiation (as with the optical pyrometer), and changes in electromotive force (as in a thermocouple-potentiometer system). Since all of these changes can be measured with reasonable accuracy, any of them may be used for relative temperature measurements.

In order to establish a usable temperature scale it is first necessary to set two fixed points. In the centigrade scale, 100 and 0 are taken as the boiling and freezing points of water at atmospheric pressure. In the Fahrenheit scale, body temperature was originally assumed to be 100, and the lowest temperature attainable with a mixture of ice and salt was thought to be 0. Translation to the more accurately determined boiling and freezing points of water at 212 and 32 F (14.7 lb per square inch atmospheric pressure) have resulted in a reevaluation of the original fixed points and their reassignment to other values on the scale.

Although all standard measuring devices show reasonable agreement between the two fixed points, extrapolation to very high or very low temperatures beyond these had led to many difficulties. The physical property incorporated in the device, such as change of volume, does not change equally for a definite temperature increment at different parts of the scale. Furthermore, this deviation from the linear varies between different types of temperature-measuring instruments. Hence some standard must be established that should be independent, if possible, of the physical properties of any substance.

After Carnot conceived his cycle, William Thomson (Lord Kelvin) incorporated this principle in the definition of a temperature scale entirely independent of the properties of any substance. This *thermodynamic temperature scale*, in the modified form developed later by Kelvin, proposed, first, that there be a Carnot-cycle system operating between two temperatures T_0 and T_1 and, second, that between these temperatures there be a series of Carnot engines, the first receiving Q_1 heat units and rejecting Q_2 heat units, the second receiving these Q_2 heat units and rejecting Q_3 heat units, and so on. This series is illustrated in Fig. 16.1. If these Carnot cycles are so chosen that the quantities of work performed

by each cycle are equal, then

$$(Q_1 - Q_2) = (Q_2 - Q_3) = (Q_3 - Q_4) = \dots$$

and $\eta_1 = \frac{Q_1 - Q_2}{Q_1} = \frac{T_1 - T_2}{T_1}, \quad \eta_2 = \frac{Q_2 - Q_3}{Q_2} = \frac{T_2 - T_3}{T_2}$

Then $\frac{Q_1 - Q_2}{T_1 - T_2} = \frac{Q_2 - Q_3}{T_2 - T_3} = \frac{Q_3 - Q_4}{T_3 - T_4} = \dots$

or $\frac{Q_1}{T_1} = \frac{Q_2}{T_2} = \frac{Q_3}{T_3} = \dots$

Therefore, if the number of equal Carnot-cycle work quantities in the temperature interval between melting ice and condensing water vapor

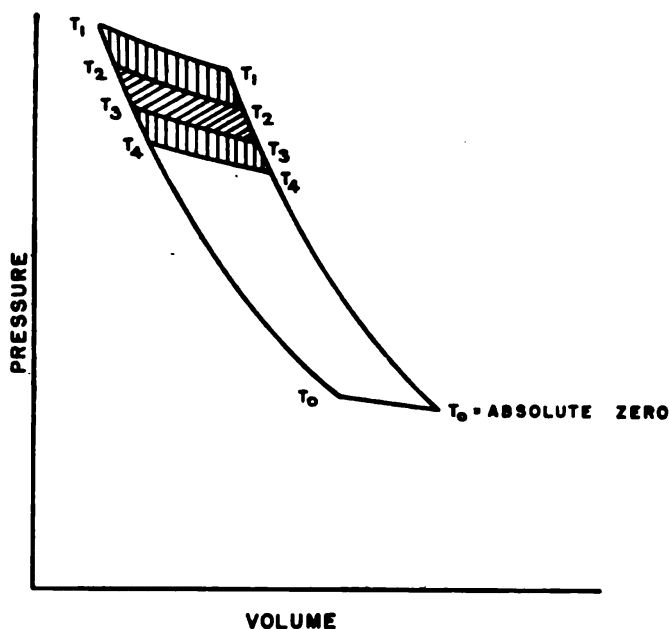


Fig. 16.1. Thermodynamic temperature scale based on series of Carnot heat engines.

is fixed at 100 units, a temperature scale has been established which is independent of the physical properties of any substance. Such a scale is called the *Kelvin scale* (K) or the *absolute centigrade scale*. If the number of work quantities in the temperature interval between melting ice and boiling water is taken as 180 units, the resulting temperature scale is called the *Rankine* or *absolute Fahrenheit scale*. Absolute zero in either case is the lower limit of temperature at which the internal energy of the last Carnot engine has been completely expended in performing external work.

Of course such a temperature scale is dependent upon the Carnot cycle for its definition—a cycle that can be approached but never attained in practice. The actual temperature-measuring instrument most closely approaching such a thermodynamic temperature scale in its reading is

the constant-volume gas thermometer. Such an instrument, shown in Fig. 16.2, is based upon Charles' law stating that the pressure of a perfect gas varies directly with the absolute temperature if the volume is kept constant. With hydrogen as the gas the pressure usually employed at the melting point of ice is 100 cm of mercury. Since it is difficult to find a container impervious to hydrogen at high temperatures, nitrogen is commonly used for such measurements. Helium is generally employed for very low temperature measurements because of its low liquefaction temperature.

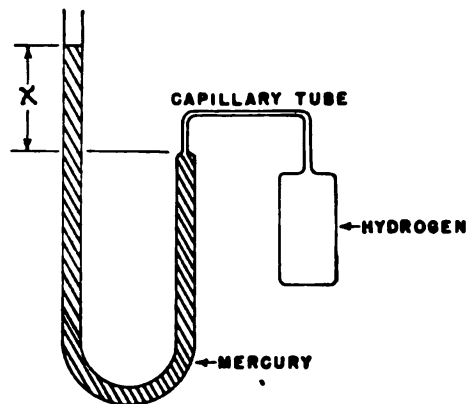


Fig. 16.2. Constant-volume gas thermometer.

Complete conformity of the readings obtained on a constant-volume gas thermometer with those of the thermodynamic temperature scale is dependent upon the use of a perfect gas. Although there are no such actual gases, the degree of deviation from the perfect can be obtained

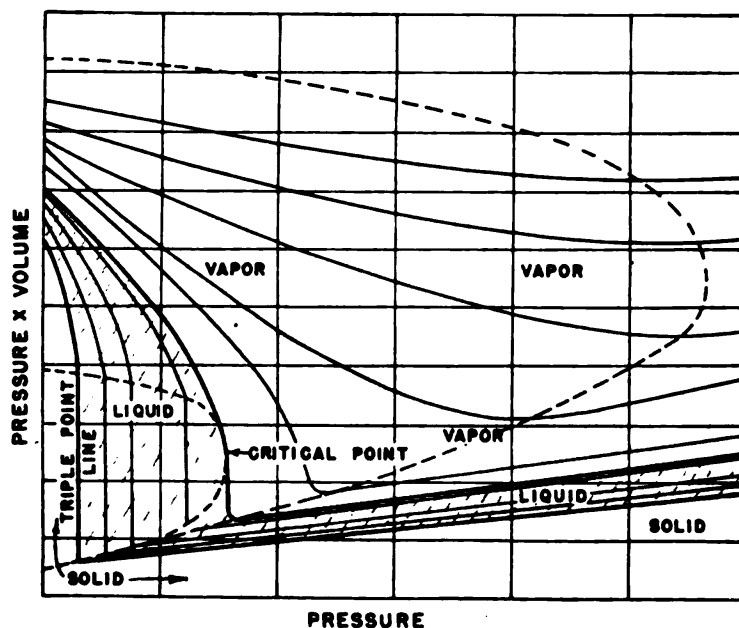


Fig. 16.3. Isotherms for a typical gas plotted on PV - P coordinates.

either (a) by determinations of the cooling or heating effect of an actual gas upon expansion through a porous plug (§16.5) or (b) by determinations of the degree of deviation of the gas from Boyle's law. For a perfect gas, $pV = a$ constant, but for a real gas, such as carbon dioxide (see Fig. 16.3), this statement is far from accurate. By establishing corrections

for such deviations from the perfect it is also possible to establish corrections for the constant-volume thermometer readings to bring them into close conformity with the thermodynamic temperature scale. Absolute zero on the thermodynamic scale coincides with absolute zero predicted experimentally with the gas thermometer.

Absolute zero temperature can be predicted by first determining the change in volume of a perfect gas with a change in temperature at constant pressure. From this change the number of degrees drop in temperature required to reduce the volume to zero can be calculated. Since a perfect gas would contract $1/459.69$ of its volume when lowered 1 F at 0 F, absolute zero is -459.69 F.

In 1927 the Seventh General Conference of Weights and Measures, including representation from 31 nations, unanimously adopted an International temperature scale based upon a series of fixed points accurately located by gas-thermometer determinations. This action was taken because the gas thermometer is a difficult instrument to use practically and, further, because the as yet partially unsolved experimental difficulties involved in the practical attainment of a thermodynamic scale made it expedient. Such a series of fixed points enables the calibration of a wide range of temperature measuring instruments with reasonable accuracy. It was further stipulated that the thermodynamic centigrade scale should be recognized as the fundamental scale and that the "International scale should conform . . . as closely as possible with the present knowledge." It is intended that revisions be made in this scale from time to time as additional information is evolved.

The fixed points set for the International temperature scale were based upon the temperature of equilibrium (a) between liquid and gaseous oxygen (90.19 K), (b) between ice and air-saturated water (273.16 K), (c) between liquid water and its vapor (373.16 K), (d) between liquid sulfur and its vapor (717.76 K), (e) between liquid silver and solid silver (1233.7 K), and (f) between liquid gold and solid gold (1336 K). All points were established at essentially standard atmospheric pressure. It is evident that this range of fixed temperatures does not extend down to extremely low temperatures. However, many low-temperature fixed points have been established with reasonable accuracy, although as yet there is no international agreement or even complete agreement among various investigators as to the exact values. Acceptable values for some of these fixed points at low temperatures are shown in Table 16.1. The triple point is defined in §3.14, and a transition point may be described as a thermodynamic or physical modification that occurs at a definite temperature. The temperature at which superconductivity occurs would be one such modification.

16.3. Production of Low Temperatures by Salt-Ice Mixtures. The establishment of an acceptable temperature scale for the accurate meas-

urement of low temperatures has constituted one of the basic problems in cryogenics. Of equal or greater importance, of course, is the development of methods by which low temperatures may be attained.

TABLE 16.1
TEMPERATURES OF FIXED POINTS BELOW THE OXYGEN POINTS²

<i>Fixed Point</i>	<i>Temperature, K</i>
Oxygen, boiling point.....	90.2
Carbon monoxide, boiling point.....	81.7
Tetrafluoromethane, transition point.....	76.2
Carbon monoxide, triple point.....	68.1
Oxygen, triple point.....	54.3
Oxygen, transition point.....	43.8
Nitrogen, transition point.....	35.6
Neon, boiling point.....	27.1
Normal hydrogen, boiling point.....	20.4
Normal hydrogen, triple point.....	14.0
Helium, boiling point.....	4.2
Helium, transition (γ).....	2.2

One of the earliest recorded methods of artificially obtaining temperatures below those of ice, and even now one of the simplest, is to place ice in contact with salts, such as ammonium sulfate or sodium chloride. The salt, when in contact with the ice, forms a salt solution with the transformation requiring the absorption of heat. This quantity of heat is equivalent to the latent heat of the ice minus the heat of solution and, if the mixture is thermally insulated, is supplied by energy contained in the mixture itself. The formation of the saline solution dissolves more ice, and this process continues until a minimum temperature is reached, depending upon the salt used. The equilibrium diagram for ice and ammonium sulfate is shown in Fig. 16.4. In this case contact between the salt and ice forms a solution of 42 per cent ammonium sulfate by weight. More ice is then dissolved and the temperature is lowered until the mass reaches a minimum temperature of 19.1 C (-2.4 F) and a concentration of 38 per cent.

16.4. Production of Low Temperatures by the Expansion of Gases. Gases may be used to produce refrigeration without a change of phase in two different ways: first, by the isentropic expansion of the gas, theoretically reversibly, against a restraining force such as a piston; and second, by the free, irreversible expansion of the gas from a comparatively high pressure through an orifice or other restriction to a lower pressure. Each method will be discussed in turn.

Isentropic Expansion of Gas. This is the method of refrigeration employed in the air refrigeration cycle discussed in §6.2 and at present

² Adapted from Hoge, Harold J., "Practical Temperature Scale Below the Oxygen Point and a Survey of Fixed Points in This Range," in *Temperature, Its Measurement and Control in Science and Industry*, American Institute of Physics. New York: Reinhold Publishing Corporation, 1941, p. 152.

utilized in some aircraft refrigeration systems. The thermodynamic relationships for this process are given in Table 3.1, and the drop in temperature corresponding to a drop in pressure may be found from the equation

$$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2} \right)^{\frac{\gamma-1}{\gamma}} \quad (16.1)$$

Although a greater lowering of the temperature of a gas may be obtained by such an expansion rather than by the irreversible adiabatic expansion

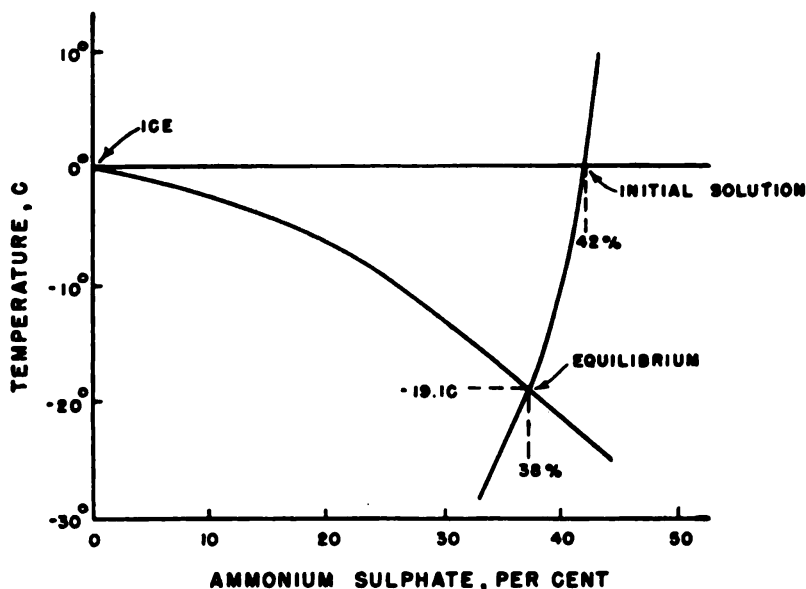


Fig. 16.4. Ice-ammonium sulphate equilibrium diagram.

of the gas, practical difficulties limit the use of this process to comparatively high temperatures. A moving expansion mechanism, such as an engine, may be subject to lubrication difficulties when directly exposed to the rapidly lowering temperatures. Irreversible expansion eliminates this problem and simplifies the equipment necessary but does so with a marked lowering of the efficiency.

Joule-Thomson or Irreversible Expansion of Gas. The free, irreversible expansion of a perfect gas results in no change in temperature. The actual drop in temperature experienced by most gases upon free expansion between two pressures may be used as a measure of the degree of imperfection of that gas and also as a means of refrigeration. The change in temperature with drop in pressure at constant enthalpy is termed the Joule-Thomson coefficient,

$$\left(\frac{\partial T}{\partial p} \right)_H = \mu, \quad (3.13)$$

which varies with both the temperature and pressure of the gas. For each gas there is an inversion point at which this coefficient is zero; above

this point there is an increase in temperature with drop in pressure, and below this point the reverse is true. This point, however, is not unique, since μ varies with pressure as well as temperature. Inversion temperatures for several common gases at atmospheric pressure are shown in Table 16.2.

TABLE 16.2
TRIPLE-POINT, CRITICAL, AND INVERSION CONSTANTS

Substance	Triple-Point Constants		Critical Constants		Maximum Inversion Temp., F
	Temp., F	Press, mm Hg	Temp., F	Press, Atm.	
Helium.....	-455.8	38.65 (Lower triple point)	-450.2	2.26	- 418
Hydrogen.....	-434.4	51.4	-399.8	12.8	- 108
Neon.....	-415.5	325.	-379.3	25.9	
Oxygen.....	-361.1	2.	-181.8	49.7	
Nitrogen.....	-345.6	96.4	-232.8	33.5	658
Argon.....	-308.6	512.2	-187.7	48.0	842
Carbon dioxide.....	- 69.9	3880.	88.0	73.	2246
Water.....	32.018 (Ice I, liquid, vapor)	4.579	705.5	218.5	
Air.....			-220.3	37.2	626

The factors affecting the magnitude of the Joule-Thomson coefficient may best be defined by the theoretical equations for the porous-plug experiment. Since the quantity of dU in the simple energy equation (Chapter 3) is an exact differential, this equation may be written

$$\begin{aligned} du &= dQ - p \, dv \\ &= T \cdot ds - p \cdot dv \end{aligned}$$

However,
and therefore

$$\begin{aligned} d(pv) &= p \cdot dv + v \cdot dp \\ d(u + pv) &= T \cdot ds + v \cdot dp \end{aligned}$$

since the sum of two quantities that are exact differentials is also an exact differential. Because the quantity $(u + pv)$ is, by definition, enthalpy,

$$dh = T \cdot ds + v \cdot dp \quad (16.2)$$

However,

$$ds = \left(\frac{\partial s}{\partial T} \right)_p dT + \left(\frac{\partial s}{\partial p} \right)_T dp,$$

and substitution of this equation in equation 16.2 results in

$$dh = T \left(\frac{\partial s}{\partial T} \right)_p dT + T \left(\frac{\partial s}{\partial p} \right)_T dp + V \cdot dp \quad (16.3)$$

The partial derivative of equation 16.3 with respect to p and with T

constant gives the equation

$$\left(\frac{\partial h}{\partial p}\right)_T = T \left(\frac{\partial s}{\partial p}\right)_T + V$$

and the partial derivative of equation 16.3 with respect to p with h constant results in the equation

$$\left(\frac{\partial h}{\partial p}\right)_h = T \left(\frac{\partial s}{\partial T}\right)_p \left(\frac{\partial T}{\partial p}\right)_h + T \left(\frac{\partial s}{\partial p}\right)_T + V = 0$$

By combination of these two equations,

$$\begin{aligned} T \left(\frac{\partial s}{\partial T}\right)_p \left(\frac{\partial T}{\partial p}\right)_h &= - \left[T \left(\frac{\partial s}{\partial p}\right)_T + V \right] = - \left(\frac{\partial h}{\partial p}\right)_T \\ &= - \left(\frac{\partial U}{\partial p}\right)_T - \left[\frac{\partial(pV)}{\partial p} \right]_T \end{aligned} \quad (16.4)$$

However, since it may be proved that

$$C_p = T \left(\frac{\partial s}{\partial T}\right)_p$$

$$\text{then} \quad C_p \left(\frac{\partial T}{\partial p}\right)_h = - \left(\frac{\partial U}{\partial p}\right)_T - \left[\frac{\partial(pV)}{\partial p} \right]_T \quad (16.5)$$

In the Joule-Thomson or porous-plug experiment the quantity observed experimentally is the change in temperature per unit pressure change or the quantity $\left(\frac{\partial T}{\partial p}\right)_h$. From equation 16.5 it may be seen that this, the Joule-Thomson coefficient, is dependent upon two competing quantities. The first of these, $\left(\frac{\partial U}{\partial p}\right)_T$, or the change in internal energy per unit change in pressure, is usually negative, but the second of these, $\left[\frac{\partial(pV)}{\partial p} \right]_T$ represents the deviation of the gas from the perfect-gas laws and may be either positive or negative. An illustration of the variation that may be expected in this term is shown in Fig. 16.3, in which pV is plotted against p for a typical substance. When the right-hand member of equation 16.5 is equal to zero, this defines the conditions for the *inversion temperature* or the temperature and pressure conditions under which expansion results in neither heating nor cooling of the gas. Under these conditions,

$$\left(\frac{\partial U}{\partial p}\right)_T = \left[\frac{\partial(pV)}{\partial p} \right]_T \quad (16.6)$$

Above these conditions expansion will result in heating, and below these conditions expansion will result in cooling.

The values of the Joule-Thomson coefficients are positive for all gases under high temperatures and pressures. Under these conditions, therefore, throttling results in a warming of the gas. The inversion temperature for most common gases (see Table 16.2) are above those of temperatures normally encountered. Therefore, if throttling occurs at ordinary temperatures and from high pressures that are not too high, a cooling effect results. However, hydrogen and helium behave at ordinary temperatures as most gases do at high temperatures, since their inversion temperatures are -108°F and -418°F , respectively. Therefore, if throttling of these gases is to be used to accomplish refrigeration, it is first necessary to cool them below their inversion temperatures by some other means. Figure 16.5 shows the temperature-entropy chart for air and may be used to determine the Joule-Thomson coefficients for this gas by reference to the temperature and pressure changes that occur along constant-enthalpy lines.

EXAMPLE 16.1. Air at 80°F and 80 psia is expanded to a pressure of one atmosphere. Determine the final temperature of the air if (a) expansion occurs adiabatically against a restraining force and (b) expansion occurs irreversibly through a throttling valve. In case (b) determine the average value of the Joule-Thomson coefficient.

SOLUTION:

$$(a) \quad T_1 = 460 + 80 = 540^\circ\text{F abs}$$

$$p_1 = 80 \text{ lb per sq in. abs}$$

$$p_2 = 14.7 \text{ lb per sq in. abs}$$

$$\gamma = 1.41$$

$$T_2 = \frac{540}{\left(\frac{80}{14.7}\right)^{\frac{1.41-1}{1.41}}} = -330^\circ\text{F abs}$$

$$t_2 = -130^\circ\text{F}$$

This same answer may be obtained graphically on Fig. 16.5 by expanding isentropically from the initial to the final pressure.

(b) Graphical expansion along the constant enthalpy path (Fig. 16.5) from the initial conditions of 80°F and 80 lb per square inch absolute to the final condition of 14.7 lb per square inch absolute indicates a final temperature of 75°F . From an examination of the temperature-entropy chart it is evident that a much greater temperature drop could be obtained by this means if the expansion originated at a higher pressure and a lower temperature.

The average value of the Joule-Thomson coefficient is

$$\mu = \left(\frac{\Delta T}{\Delta p}\right)_h = \frac{5}{65.3} = 0.076$$

16.5. Production of Low Temperatures by Evaporation. The absorption of heat accompanying the transformation of a liquid to a vapor is utilized not only in the standard vapor-compression refrigeration cycle but also as a means of producing very low temperatures. The container

for the evaporating liquid should be insulated from sources of heat other than the liquid itself, and the vapor formed in the process should be removed as rapidly as it is formed. By this process the triple point may be reached and the substance completely solidified. Hydrogen was solidified (14.0 K) thus as early as 1899, and helium remained the only gas not solidified until 1926.

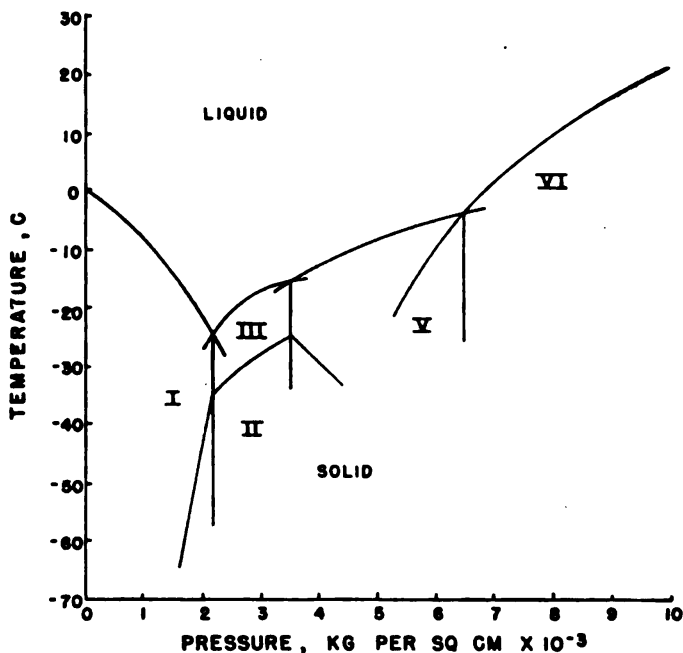


Fig. 16.6. Temperature-pressure diagram for liquid and solid water. From *Journal Franklin Institute*, Vol. 177, 1914, pp. 315-332.

Prior to 1926 many unsuccessful attempts had been made to solidify helium. Onnes at the University of Leiden had succeeded in lowering the temperature to approximately 0.82 K by surrounding the helium with boiling hydrogen and lowering the pressure to 0.013 mm of mercury. Because no solid helium resulted, it was concluded that its solidification was impossible.

Keesom,³ also at the University of Leiden, tried a different approach to the problem in 1926. Bridgman⁴ had found that by subjecting some liquids to extremely high pressures it is possible to raise the transformation temperature and that many of the physical properties undergo marked changes. For example, as indicated in Fig. 16.6, ice was shown to possess at least five allotropic forms. If ice at -20°C (4°F) is subjected to increasing pressures up to 7000 kg per square centimeter, it can be made to pass through four allotropic forms and change from a solid to a liquid

³ Cork, J. M., *Heat*. New York: John Wiley & Sons, Inc., 1937, p. 245.

⁴ Bridgman, J., *J. Franklin Institute*, Vol. 177 (1914), pp. 315-332.

and back to a solid again. By using this same approach and subjecting liquid helium to the high pressure of 150 atm, Keesom succeeded in attaining solid helium.

The attainment of extremely low temperatures is usually accomplished by the evaporation of liquefied gases, in insulated containers and sometimes at reduced pressures. By this means 90 K may be reached with liquid air, 54.3 K with liquid oxygen, 35.6 K with liquid nitrogen, 14 K with liquid hydrogen, and temperatures below 1 K with liquid helium.

16.6. Production of Low Temperatures by Magnetic Cooling. Helium is the gas that approaches the ideal most nearly. The lowest temperature reached by boiling its liquid under the lowest attainable pressures is 0.71 K, and lower temperatures must be obtained by other means. The lowest recorded temperatures have been obtained through the demagnetization of certain paramagnetic salts previously cooled by liquid helium and subjected to a strong magnetic field.

All substances may be divided into two classes with respect to their magnetic properties. Those repelled by a magnetic pole are diamagnetic, and those attracted, such as iron, are paramagnetic. Some of the paramagnetic salts, such as gadolinium sulfate, have been found to be best suited for obtaining low temperatures by their adiabatic demagnetization. If such salts are precooled to a very low temperature so that any thermal motion of the molecules will be at a minimum, the molecules may be considered as elementary magnets that will be aligned when subjected to a strong magnetic field. If the substance is then demagnetized, work is performed without external heat exchange. Although this work is not mechanical, the result is adiabatic cooling.

In practice the procedure of cooling magnetically is accomplished in four steps.⁵ First, a paramagnetic salt is cooled by surrounding it with liquid helium boiling under reduced pressure. Second, a magnetic field on the order of 25,000 gauss is applied to the salt, still surrounded by the boiling helium, and the evolved heat is absorbed without change in temperature. Third, the helium gas is removed and the substance is thermally isolated at a temperature below 1 K and under the stress of a strong magnetic field. Fourth, adiabatic demagnetization, comparable to adiabatic expansion of a gas, further lowers the temperature when the magnetic field is reduced to zero. By this means temperatures as low as 0.004 K have been reached, and it is believed temperatures as low as 0.001 K are attainable. Such temperatures cannot be measured by ordinary means but can be determined by measuring the magnetic susceptibility and applying Curie's law, which states that this susceptibility is inversely proportional to the absolute temperature. Figure 16.7 shows

⁵ Debye, Peter, "The Magnetic Approach to the Absolute Zero of Temperature." *American Scientist*, Vol. 32, No. 4 (October, 1944), p. 229.

schematically the arrangement of materials used in approaching absolute zero magnetically.

It has been proposed that further cooling be accomplished by aligning and then demagnetizing the nuclei of the atoms as well as the paths of the electrons. It is predicted that temperatures within one one-millionth of a degree of absolute zero may be reached in this manner.

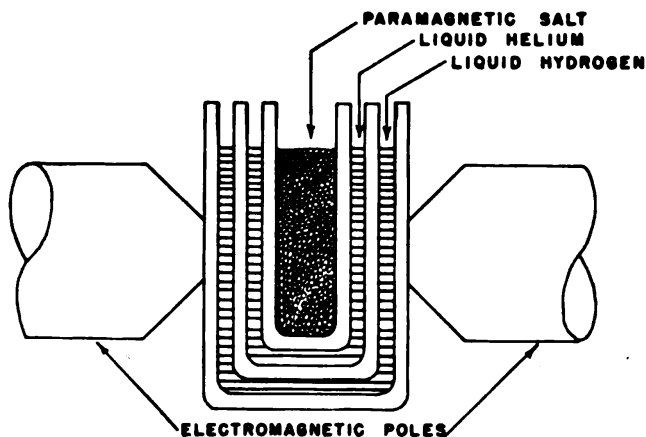


Fig. 16.7. Schematic arrangement for magnetic cooling of a paramagnetic substance.

16.7. Production of Low Temperatures by Vapor-Compression Refrigeration. The use of vapor-compression refrigeration for the production of low temperatures is limited inherently by the solidification temperature of the refrigerant (Table 6.5), the evaporation pressure corresponding to the desired temperature, and the difficulties encountered in the operation of any mechanical equipment at very low temperatures. At -100°F the corresponding absolute pressure for ammonia is 1.24 psia, at -155°F the pressure for Freon-22 is 0.19901 psia, and at -100°F the pressure for propane is 3.2 psia. It is difficult to operate compressors efficiently under such low pressures when large volumes of gases must be pumped. Many refrigerants may be used for low-temperature refrigeration, but Freon-12 and Freon-22 are by far the most widely applied. Although for small loads single-stage compression has been used with evaporator temperatures as low as -50 to -60°F , economy of operation usually limits such operation to conditions above -20°F . Two-stage Freon-12 systems are adaptable to evaporator temperatures as low as -75°F and three-stage Freon-12 systems to -90°F . Freon-22 systems operate satisfactorily at temperatures 10 to 15 $^{\circ}\text{F}$ lower. Figure 16.8 shows the single-stage-compressor displacement requirements per ton of refrigeration for Freon-12 and Freon-22 at different suction temperatures, and Fig. 16.9 approximates performance curves for three stages of compression for the same refrigerants. The theory of stage compression was covered in §13.5.

Recently several large low-temperature refrigeration installations have been designed to use four-stage centrifugal compressors with Freon-12

as the refrigerant. The National Advisory Committee's Aircraft Engine Research Laboratory at Cleveland, the world's largest low-temperature refrigeration plant at the time of installation, operates in this manner. Some of the details are presented in §17.4. One inherent advantage in the use of centrifugal compression equipment for low temperatures is the reduction in lubrication difficulties.

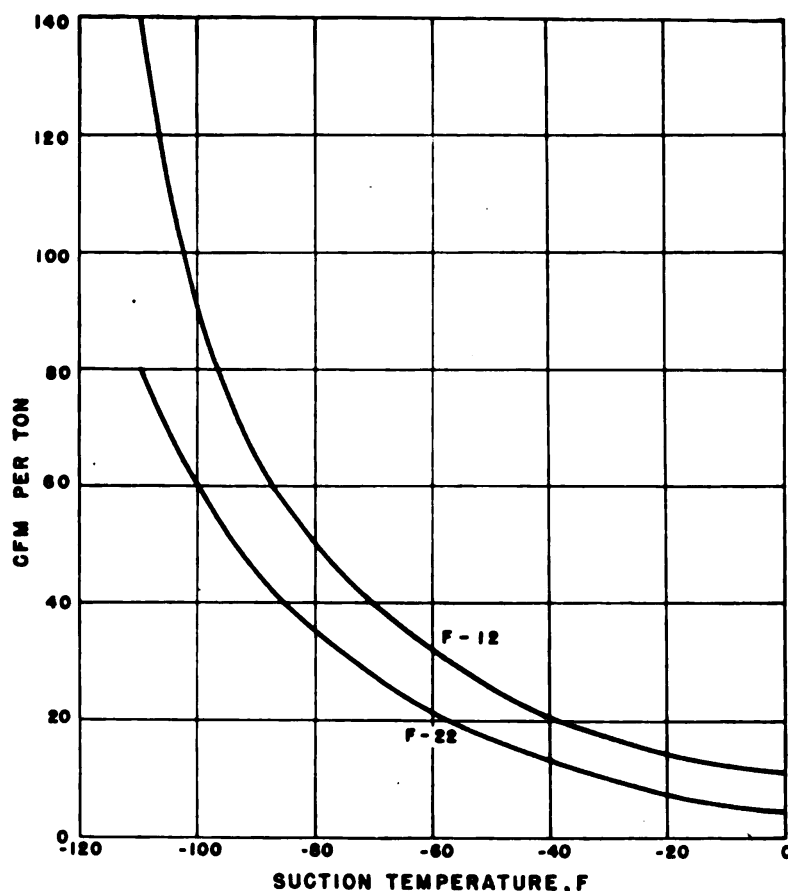


Fig. 16.8. Refrigerant displacement per ton at low temperatures. Approximate net displacement in cfm per ton. From *Refrigerating Data Book Applications' Volume*, 2nd ed., American Society of Refrigerating Engineers, 1946, p. 415.

If vapor-compression systems are to be used for the production of low temperatures, the common alternative to stage compression is the *cascade system*, in which a series of refrigerants with progressively lower boiling points is used in a series of single-stage units. The evaporator of the first system operating at the highest temperature is used to cool the condenser of the second system, the evaporator of this unit is used to cool the condenser of the third system, and so on. Thus each refrigerant circuit is comparatively simple and is a system in itself, and each refrigerant can be chosen that operates best within the required comparatively narrow temperature and pressure range.

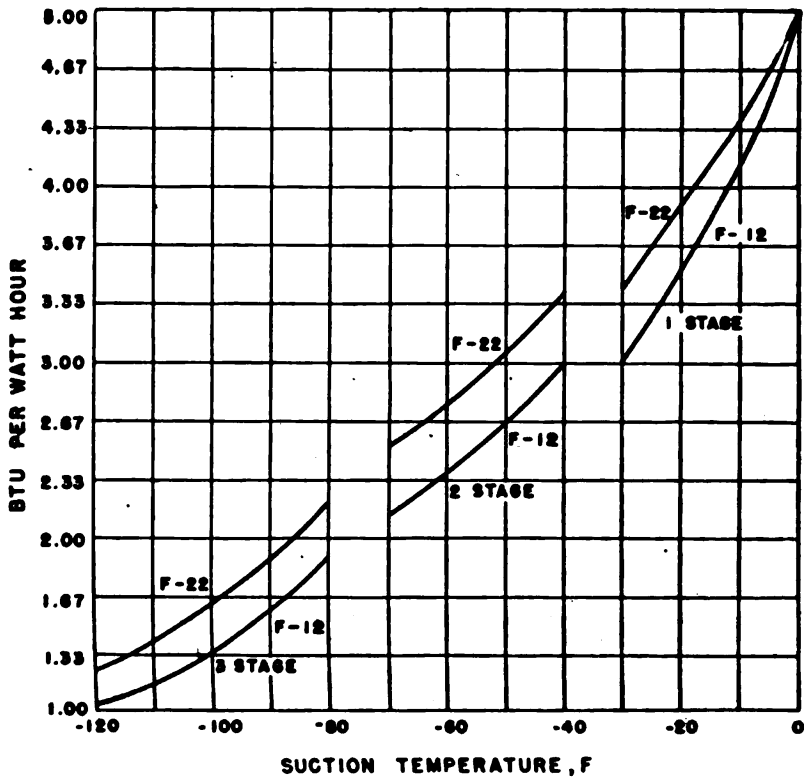


Fig. 16.9. Performance of condensing units under sub-zero conditions. From *Refrigerating Data Book Applications' Volume*, 2nd ed., American Society of Refrigerating Engineers, 1946, p. 416.

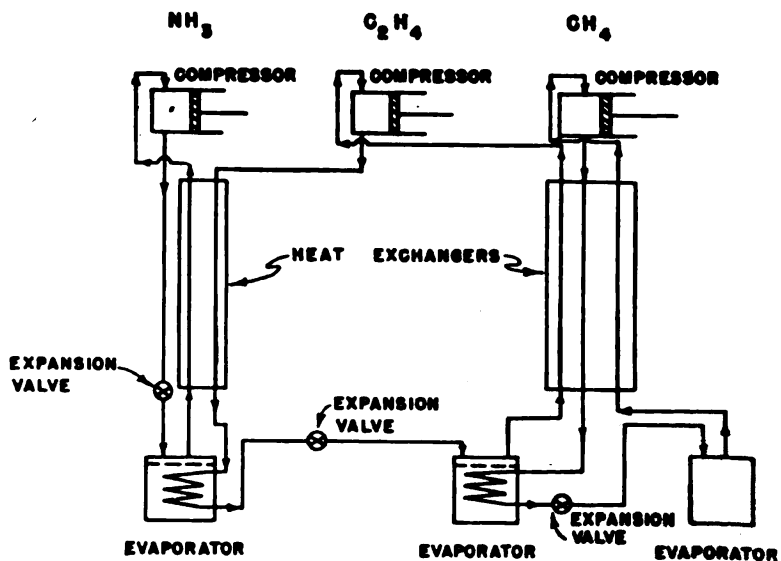


Fig. 16.10. Cascade system using ammonia, ethylene, and methane.

The cascade system was first used by Pictet in 1877 for the liquefaction of oxygen. This installation employed as intermediate refrigerants sulfur dioxide and carbon dioxide. Among other combinations used since that time are methyl chloride, ethylene and oxygen or nitrogen, and ammonia, ethylene, methane, and nitrogen. Figure 16.10 shows diagrammatically a cascade system using ammonia, ethylene, and methane as refrigerants.

EXAMPLE 16.2. A cascade refrigeration system is designed to supply 1 ton of refrigeration at an evaporator temperature of -80°F and a condenser temperature of $+80^{\circ}\text{F}$. The load at -80°F is absorbed by a unit using Freon-22 as the refrigerant and is rejected to a condenser at -10°F . This condenser is cooled by a unit using Freon-12 as the refrigerant and operating between a -20°F evaporating temperature and 80°F condensing temperature. The refrigerant leaving the Freon-12 condenser is subcooled to 70°F , but there is no subcooling of the Freon-22 refrigerant. The gas leaving both evaporators is dry and saturated, and compression is adiabatic. Neglect all compression and volumetric losses. Determine: (a) compression ratio for both units; (b) pounds of refrigerant circulated per (ton refrigeration)(min) for both units; (c) theoretical horsepower per ton of refrigeration for both units; (d) theoretical displacement required per ton of refrigeration for both units; (e) c.p. for both units; (f) discharge temperature at end of compression for both units; (g) total theoretical horsepower requirements. (h) If a single Freon-22 compressor is operated between -80°F evaporating and $+80^{\circ}\text{F}$ condensing, determine the same values, assuming 10°F subcooling.

SOLUTION:

(a) Compression ratio:

$$\text{For F-22 unit, } \frac{31.29}{4.787} = 6.54$$

$$\text{For F-12 unit, } \frac{98.76}{15.28} = 6.46$$

(b) Pounds of refrigerant circulated per (ton)(min):

$$\text{For F-22 unit, } \frac{200}{95.68 - 7.96} = 2.28 \text{ lb per (ton)(min)}$$

$$\text{For F-12 unit, } \frac{200}{75.87 - 23.90} = 3.85 \text{ lb per (ton)(min)}$$

(c) Theoretical horsepower per ton:

$$\text{For F-22 unit, } 4.717 \left(\frac{114.5 - 95.68}{95.68 - 7.96} \right) = 1.01 \text{ hp per ton}$$

$$\text{For F-12 unit, } 4.717 \left(\frac{90.81 - 75.87}{75.87 - 23.90} \right) = 1.36 \text{ hp per ton}$$

(d) Theoretical displacement per (ton refrigeration)(min):

$$\text{For F-22 unit, } (2.28)(9.650) = 22.0 \text{ cu ft per (ton)(min)}$$

$$\text{For F-12 unit, } (3.85)(2.474) = 9.5 \text{ cu ft per (ton)(min)}$$

(e) Coefficient of performance:

$$\text{For F-22 unit, } \frac{95.68 - 7.96}{114.5 - 95.68} = 4.67$$

$$\text{For F-12 unit, } \frac{75.87 - 23.90}{90.81 - 75.87} = 3.48$$

(f) Discharge temperature at end of compression:

$$\text{For F-22 unit} = 62 \text{ F}$$

$$\text{For F-12 unit} = 105 \text{ F}$$

(g) Total theoretical horsepower required:

$$1.01 + 1.36 = 2.37 \text{ hp per ton}$$

(h) Total theoretical displacement required:

$$22.0 + 9.5 = 31.5 \text{ cu ft per (ton)(min)}$$

(i) Single F-22 unit:

$$\text{Compression ratio} = \frac{159.7}{4.787} = 33.4$$

$$\begin{aligned} \text{Refrigerant circulation rate} &= \frac{200}{95.68 - 30.99} \\ &= 3.09 \text{ lb per (ton)(min)} \end{aligned}$$

$$\begin{aligned} \text{Theoretical horsepower required} &= 4.717 \left(\frac{134.96 - 95.68}{95.68 - 30.99} \right) \\ &= 2.86 \text{ hp per ton} \end{aligned}$$

$$\begin{aligned} \text{Theoretical displacement required} &= (3.09)(9.650) \\ &= 29.8 \text{ cu ft per (ton)(min)} \end{aligned}$$

$$\begin{aligned} \text{Coefficient of performance} &= \frac{95.68 - 30.99}{134.96 - 95.68} \\ &= 1.65 \end{aligned}$$

$$\text{Compressor discharge temperature} = 210 \text{ F}$$

It should be noted that the inclusion of corrections for volumetric and compression losses will affect the single-stage calculation much more unfavorably than the cascade system.

16.8. Liquefaction of Air. The liquefaction of air has become an important industrial process not only for the production of liquid air itself but also in the separation of oxygen, nitrogen, hydrogen, helium, and many of the rare gases by fractional distillation. It appears probable that considerable quantities of liquid oxygen may be used in the future for synthesizing engine fuels by the treatment of natural gas and coal and that it may find uses in the metallurgical industry and in the manufacture of heating gas.

Cascade systems of refrigeration have been used satisfactorily for the liquefaction of gases, but more frequently the methods used have applied either Joule-Thomson expansion, expansion against a restraining force,

or both. The simplest of these is the *Linde or Hampson system* shown in Fig. 16.11. Here air is compressed between points 1 and 2 to a pressure of 100 to 200 atm and is then cooled in its passage through the heat exchanger between points 2 and 3 to a temperature of approximately -160°F . An irreversible adiabatic expansion occurs between points 3 and 4 with the air dropping to atmospheric pressure and a temperature of -312°F corresponding to the boiling point. A portion of the air, depending primarily upon the initial pressure, is liquefied; with an initial pressure of 200 atm this amounts to approximately 10 per cent. The remainder of the cold gas leaves the separator at point 5 and returns through the heat exchanger to the compressor. A simplified temperature entropy diagram representing this process is shown in Fig. 16.12.

EXAMPLE 16.3. Determine the liquid air yield for a simple Linde liquefaction system if the air is compressed to 2500 psia and cooled before expansion to -170°F .

SOLUTION. The air is expanded along a constant-enthalpy line from 2500 psia and -170°F to a final state point at 1 atm. The entropy after expansion is 0.525, and the yield of liquid air is

$$\frac{0.610 - 0.525}{0.610 - 0.0} \times 100 = 13.9\%$$

The simple Linde or Hampson system is comparatively inefficient and is used only when small quantities of liquid air are needed. The improved Linde system is somewhat more efficient. Here the simple system is modified to provide for two-stage expansion and compression, with a portion of the gas after the first expansion bled back through a heat exchanger into the discharge of the first stage of compression. Increased economy results from this arrangement because the majority of the gas undergoes only one expansion to the intermediate pressure and one compression from this intermediate pressure to the high pressure.

The *Claude liquefaction system* shown in Figs. 16.13 and 16.14 combines Joule-Thomson expansion and expansion against a restraining force. Here the air is compressed to approximately 40 atm between 1 and 2 and is then partially cooled by passage through the heat exchanger, 2-3. The air stream is then divided, about 20 per cent passing through heat exchanger 3-4 and the remaining 80 per cent passing through expander 3-5. The portion sent through the second heat exchanger is then throt-

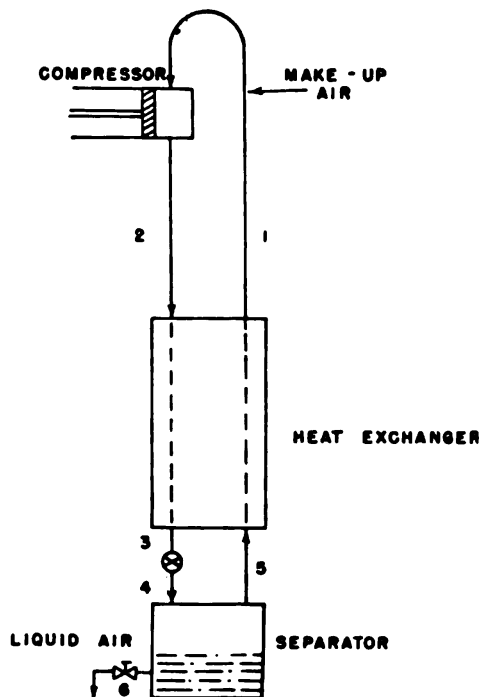


Fig. 16.11. Linde system for air liquefaction.

bled irreversibly through 4-6 into the separator at atmospheric pressure. The portion diverted through the expander enters at a temperature in

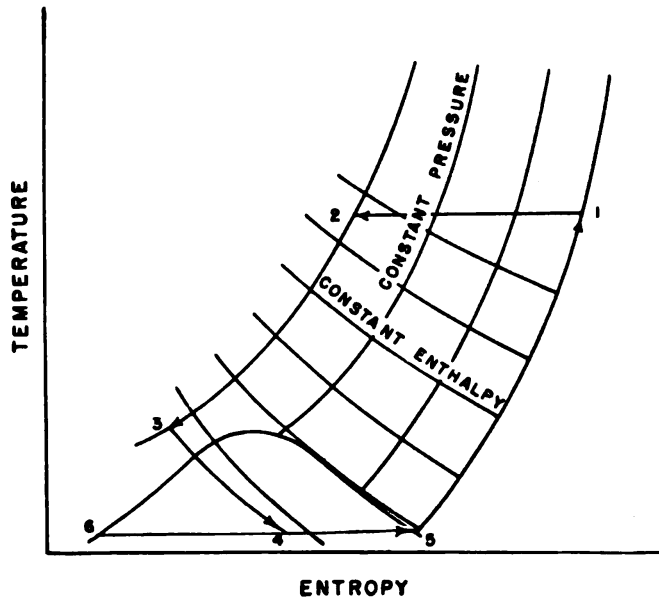


Fig. 16.12. Temperature-entropy diagram for Linde system.

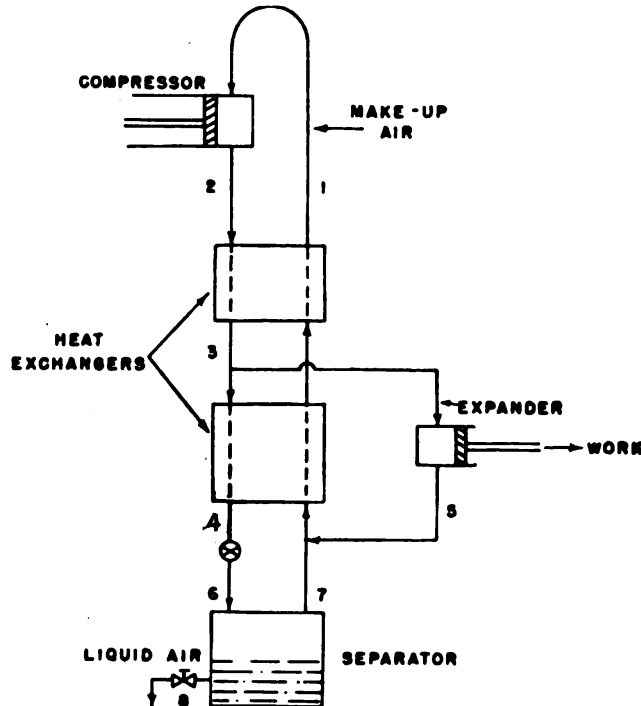


Fig. 16.13. Claude system for air liquefaction.

the order of -100°F and is then further cooled by expansion. It is then mixed with the unliquefied gas discharged from the separator at 7 and passes back through the two heat exchangers to the compressor.

pressure may be used and that chemical purification of the incoming air is unnecessary. Both ringless-piston expansion engines and turbo-expanders have been developed that may be adapted to such systems.

16.9. Manufacture of Dry Ice. Solid carbon dioxide, or dry ice, despite comparatively high manufacturing costs, has many unusual properties as a refrigerant. It is nontoxic and noncorrosive, and it is easily handled and cut. As shown in the pressure-temperature diagram of Fig. 16.15, at atmospheric pressure it sublimates directly from the solid to the

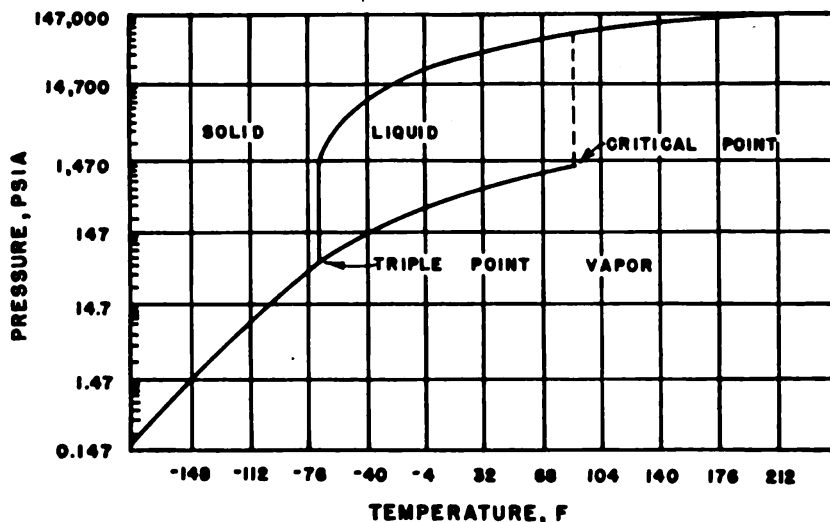


Fig. 16.15. Carbon-dioxide phase diagram.

gaseous phase at a temperature of -109°F and may therefore be used to maintain a wide range of temperatures up to those required for the preservation of unfrozen perishables. At atmospheric pressure the heat of sublimation is 246 Btu per pound, and if the gas is further raised to 32°F , an additional 27 Btu per pound of sensible heat is available. By increasing the pressure the temperature of sublimation can be raised to the triple point at -69.9°F , and above the corresponding pressure the solid passes first into the liquid and then into the gaseous phase.

The preservation of solid carbon dioxide is difficult because of its extremely low temperature. Losses may be reduced by the use of insulation, increased pressures, or both; in economical application, however, losses are unavoidable and storage must be limited to a few weeks at most.

The manufacture of carbon dioxide gas is a chemical engineering problem in itself. The heating of limestone or its treatment with acid is the principal source of the gas. Large volumes are also obtained by the controlled combustion of coke, and in recent years another source has been the process of hydrogenation in the petroleum industry. Some carbon dioxide has been obtained commercially from the fermentation of organic substances.

Carbon dioxide gas may be liquefied by compression to pressures of

900 to 1000 psi, followed by condensation in coils, which are usually cooled by water. Compression is usually accomplished by three-stage reciprocating compressors. Solidification of the liquid may be done in one of three ways: by removal of heat through auxiliary refrigeration, by removal of heat through evaporation of a portion of the liquid carbon dioxide, or by expansion of the liquid carbon dioxide to a pressure below that of the triple point. The first method has been found uneconomical because of the very low temperatures that must be attained by the external refrigeration equipment. Modifications of the second method have found

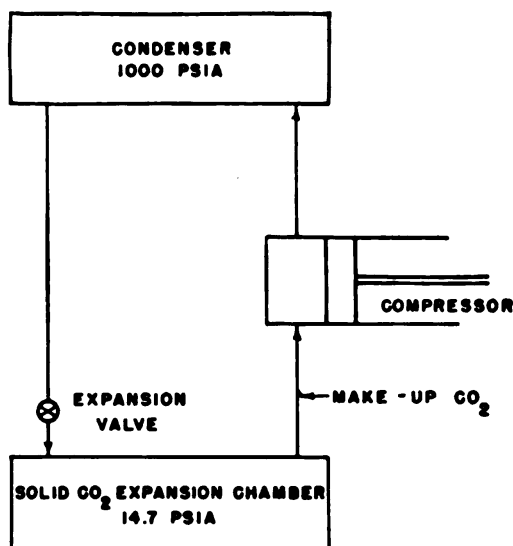


Fig. 16.16. Simplified apparatus for production of solid carbon dioxide.

limited use commercially. The third procedure, however, is the one most widely practiced for the large-scale production of dry ice.

The simplest form of equipment that could be used for the production of solid carbon dioxide by expansion of the liquid is shown in Fig. 16.16. The apparatus necessary is practically identical with that of any vapor-compression refrigeration system with the exceptions that means must be provided for the introduction of make-up carbon dioxide gas and for the removal of the carbon dioxide "snow." As expansion reduces the pressure below that of the triple point, the liquid flashes directly into gas and CO_2 snow at the discharge of the expansion valve. This snow is quite porous and must be removed from the evaporator or snow chamber and compressed by mechanical means into cakes. Although such a system as this would operate satisfactorily, the power consumption, approximately 450 hp-hr per ton of solid, would be much too high for economical production.

There are several modified forms of the system shown in Fig. 16.16 that result in a greater yield of solid and a lower unit power consumption. Stage compression and intercooling may be used; the snow chamber may

be operated at a pressure only slightly below that corresponding to the triple point instead of at atmospheric pressure; a cascade system may be used with the condenser of the carbon dioxide system maintained at 0 F by means of an ammonia, Freon, or similar cycle. Any of these modifications increases the efficiency of the basic cycle. By a combination of the cascade system, stage compression of the carbon dioxide, and operation of the snow chamber at pressures above atmospheric, it is possible

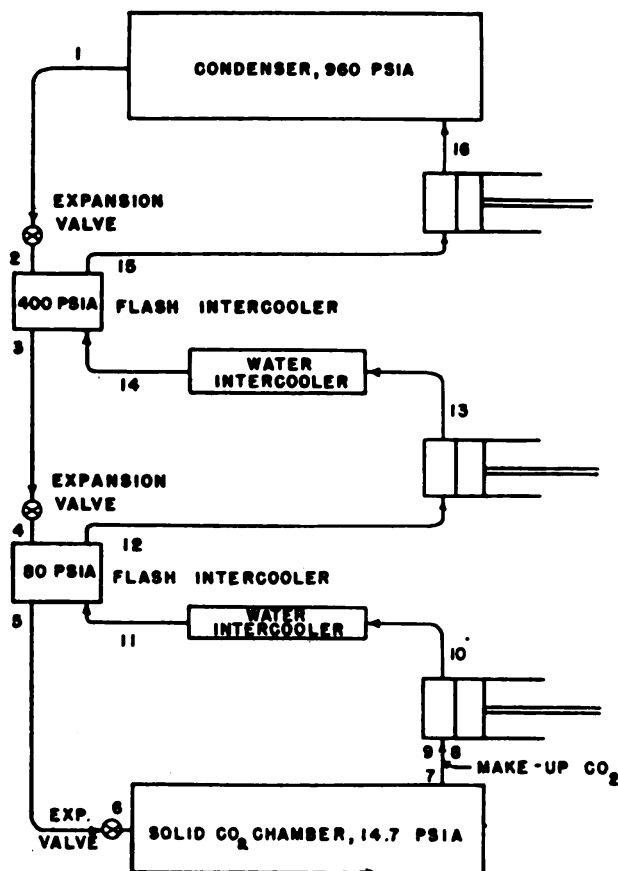


Fig. 16.17. Three-stage system for production of dry ice.

to reduce the power consumption to well below 200 hp-hr per ton of snow.

One possible modification of the elementary system is shown in Fig. 16.17. Here compression is accomplished in three stages, and both flash and water intercooling are used. The snow chamber is operated at atmospheric pressure. The flash intercoolers operate so as to result in thermal equilibrium between the partially expanded liquid refrigerant and the returning vapor.

EXAMPLE 16.4. A three-stage system for the manufacture of solid carbon dioxide operates under the following pressures: condenser, 960 psia; intermediate-stage-compressor discharge 400 psia; low-stage-compressor discharge, 80 psia; snow or solid CO₂ chamber, 14.7 psia. Both flash and water intercoolers are used between stages (see Fig. 16.17). Make-up CO₂ enters at 70 F and atmos-

pheric pressure and the gas leaving both water intercoolers is at 70 F. Calculate the theoretical horsepower per (ton solid CO₂ per hr) for operation of the system.

SOLUTION. Flash intercoolers are designed to bring the refrigerant vapor passing between compressor stages into intimate contact with the liquid refrigerant passing between expansion stages. Sufficient liquid refrigerant is flashed to remove the superheat from the discharged compressor gases. All the refrigerant vapor is drawn from the flash intercooler into the following compressor stage, and only liquid refrigerant passes from the intercooler into the expansion valve. Since under equilibrium conditions the weight of refrigerant circulated in each circuit must remain constant,

$$W_{15} = W_2, W_{14} = W_3 = W_{12} = W_4 \quad \text{and} \quad W_{11} = W_5 = W_6 = W_7 + W_8$$

The weight of make-up CO₂ must equal the weight of solid CO₂ per unit of time. Each flash intercooler must also be in thermal equilibrium. Enthalpy values based on a datum of -40 F are as follows:

$h_1 = h_2 = 73$ Btu per lb	$h_{12} = 136$ Btu per lb
$h_3 = h_4 = 27$	$h_{13} = 167$
$h_5 = h_6 = -12$	$h_{14} = 155$
$h_7 = 133$	$h_{15} = 138$
$h_8 = 169$	$h_{16} = 152$
$h_{11} = 167$	

After expansion of the refrigerant into the solid CO₂ chamber,

$$x = \frac{-12 + 113}{133 + 113} = 0.508$$

and the product yield = $1 - 0.41 = 0.59$ lb solid CO₂ per pound of refrigerant.

$$\begin{aligned} h_9 &= (0.59)(169) + (0.41)(133) \\ &= 154.3 \text{ Btu per lb} \end{aligned}$$

and, with adiabatic compression,

$$h_{10} = 194 \text{ Btu per lb}$$

The weight of refrigerant that must be circulated through the low-pressure circuit in order to produce a ton of solid CO₂ per hour is

$$W_9 = W_7 + W_8 = \frac{2000}{(60)(0.59)} = 56.5 \text{ lb per min}$$

The weight of refrigerant circulated through the intermediate pressure circuit is

$$\begin{aligned} W_{12} &= \frac{W_{11}(h_{11} - h_5)}{(h_{12} - h_4)} \\ &= \frac{(56.5)(167 - -12)}{136 - 27} = 92.6 \text{ lb per min} \end{aligned}$$

The weight of refrigerant circulated through the high pressure circuit is

$$\begin{aligned} W_{15} &= \frac{W_{14}(h_{14} - h_3)}{(h_{15} - h_2)} \\ &= \frac{92.6(155 - 27)}{138 - 73} = 182.5 \text{ lb per min} \end{aligned}$$

The required horsepower is

$$\begin{aligned} \text{hp} &= \frac{56.5(194 - 154) + 92.6(167 - 136) + 182.5(152 - 138)}{42.42} \\ &= 181.5 \text{ hp per (ton solid CO}_2 \text{ per hr)} \end{aligned}$$

16.10. Applications of Low-Temperature Refrigeration. The attainment of extremely low temperatures, within a few degrees of absolute zero, has its applications primarily in the realm of pure research. Investigations at temperatures below 1 K are carried out in studies of the residual entropy of matter. A perfect crystal will have no entropy at absolute zero, but solids with a less orderly arrangement will have entropy values corresponding to the extent of their disorder.

Low-temperature research has proved very fruitful in providing knowledge of some interesting phenomena. One of these is the superconductivity of many metallic conductors at very low temperatures. For reasons still incompletely understood, certain metals such as tin, lead, aluminum and zinc lose practically all of their electrical resistance at temperatures of one or two degrees Kelvin. Spectrum lines are usually sharpened, and the quantum levels of solids are sometimes best investigated through absorption spectra at very low temperatures. Many transition phenomena in metals occur at low temperatures. Because of the reduced amplitude of thermal vibrations, the lattice parameters of metals are best studied under these conditions. Low-temperature techniques have been applied with good results to nuclear problems. For example, helium 3, with a mass of 3 instead of 4 and occurring in the atmosphere in only 1 part per million parts of air, has been separated by cryogenic laboratory techniques and has proved important in nuclear researches. The possibilities of basic research of many physical problems at low temperatures are vast and to a large extent unexplored.

In addition to the fractional distillation of gases and the manufacture of solid carbon dioxide, there are an increasing number of commercial applications of low-temperature refrigeration. Some of these applications in the fields of medicine, metallography, and engineering are discussed in Chapter 17.

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PROBLEMS

16.1. Calculate the average Joule-Thomson coefficient for a pressure drop to atmospheric pressure from 200 atm when the initial temperature is (a) 150 F, (b) 0 F, (c) -150 F.

16.2. A simple Linde air-liquefaction system compresses dry air isothermally from 70 F and 14.7 psia to 2500 psia. If the heat exchanger is 60 per cent efficient (based on temperature change), determine (a) air temperature at entrance to expansion valve, (b) percentage of liquefied air and (c) theoretical horsepower-hours per pound of liquid air.

16.3. The suction pressure within a reciprocating compressor corresponds to that for saturated vapor at -100 F. Determine the evaporator temperature if the throttling loss across the suction valves is 3 psi and the refrigerant is (a) carbon dioxide, (b) ethylene, (c) Freon-11, (d) Freon-22. (Assume saturated vapor at evaporator discharge for an approximate solution.)

16.4. In Example 16.2 both compressors operate with 2% clearance, a 1.5 psi suction-valve pressure drop and a 2.5 psi discharge-valve pressure differential. For the Freon-22 system $n = 1.15$, and for the Freon-12 system $n = 1.10$. Solve the problem under these conditions.

16.5. A cascade refrigeration system is designed for the liquefaction of natural gas. A two-stage ethylene compressor operates between an evaporator temperature of -145 F and a condenser temperature of -10 F. A two-stage ammonia compressor operates in cascade with the ethylene unit between an evaporator temperature of -20 F and a condenser temperature of 80 F. Each two-stage unit is proportioned so that the cylinder compression ratios are equal. A stage intercooler capable of reducing the vapor temperature to 75 F is used with

the ammonia system and a flash intercooler with the ethylene system. Disregard compressor clearance and valve wire-drawing, and assume isentropic compression. Determine (a) total horsepower per ton of refrigeration, (b) piston displacement per (min)(ton refrigeration) for each compressor, and (c) coefficient of performance (c.p.) for each compressor and for the system.

16.6. Carbon dioxide, eventually to be used in the liquid or gaseous phase, is frequently shipped in solid form to eliminate the shipping of heavy storage cylinders. If 50 lb of solid CO_2 is placed in a steel cylinder of 1.2 cu ft internal volume and allowed to attain equilibrium with the surrounding air at 70 F, determine (a) internal cylinder pressure, psia; (b) weight of liquid CO_2 ; (c) weight of gaseous CO_2 ; and (d) Btu absorbed by solid CO_2 in attaining equilibrium.

16.7. Compare the cost of refrigeration per 1000 Btu supplied by (a) dry ice, (b) water ice, (c) mechanical refrigeration at 0 F suction, and (d) mechanical refrigeration at -110 F suction. Assume local costs for dry ice, water ice, and electric power. Assume condensing-unit performance shown in Fig. 16.9 and Freon-22 as the refrigerant. (In comparing cost it should be noted that approximately eight times as much refrigeration by water ice as by dry ice is required for satisfactory cooling of ordinary refrigerator cars. This relationship is partially explained by the closer proximity of load to cooling surfaces and by more efficient control.)

16.8. Fresh frozen shrimp are to be shipped by air in 7 hr from Florida to New York. The shrimp, prefrozen, are to be packed in insulated containers $4 \times 4 \times 4$ ft ($U = 0.25$). Dry ice at 14.7 psia is used as the refrigerant and is so placed in the container that the average temperature adjacent to the walls may be considered as 0 F. If the ambient temperature is 60 F, determine (a) pounds of dry ice required for shipment and (b) cost of refrigeration if dry ice is available at $3\frac{1}{2}$ cents a pound.

16.9. A three-stage system for the manufacture of solid CO_2 operates with the following pressures: condenser, 1000 psia; intermediate-stage-compressor discharge, 430 psia; low-stage-compressor discharge, 90 psia; and snow or solid CO_2 chamber, 14.7 psia. Both flash and water intercoolers are used between stages. Make-up CO_2 enters at 75 F and atmospheric pressure, and the gas leaving the water intercoolers is at 70 F. Determine (a) theoretical horsepower per (ton solid CO_2 per hr) (b) theoretical piston displacement in cubic feet per minute for each ton of solid CO_2 per hour.

16.10. The solid CO_2 system of Problem 16.9 is modified to a cascade or binary system, with a Freon-22 system replacing the high pressure CO_2 stage. Thus the two-stage CO_2 system discharges to a condenser operating at 430 psia, and this in turn is cooled by the Freon-22 evaporator operating at 49.58 psia. The Freon-22 condenser pressure is 174.5 psia. Determine (a) theoretical horsepower per (ton solid CO_2 per hr), and (b) theoretical piston displacement in cubic feet per minute for each ton of solid CO_2 per hour.

PART V
Refrigeration Applications

CHAPTER 17

Applications of Refrigeration

17.1. General. Refrigeration can serve us from the cradle to the grave. For some, the benefits start at birth in the air-conditioned delivery room and for a few in the modern incubator¹ and nursery for premature babies. Applications are then encountered and appreciated, though often indirectly, throughout life. And for some, refrigeration is even applied after death in cooling the slab vault at the city morgue. An analysis of these applications will indicate that they may be classified into one of the following three general groups:

- A. An aid to some other process
- B. A means of preservation
- C. A factor in comfort improvement

More specific classifications would include the following:

1. Storage of foods above and below freezing
2. Transportation of foods above and below freezing
3. Processing food products and beverages
4. Freezing food products
5. Industrial air conditioning
6. Comfort air conditioning
7. Ice making
8. Chemical and related industries
9. Oil refining and synthetic rubber manufacturing
10. Manufacturing and treatment of metals
11. Creation of artificial atmospheric conditions
12. Medical and surgical aids
13. Heat pump
14. Miscellaneous cooling processes

17.2. Heat Pump. "Heat pump" is the modern expression for a refrigerating system in which the heat discharged at the condenser is of prime interest and importance. The cooling effect produced by the evaporator is secondary and is not necessarily utilized. The medium being cooled serves as a heat source, and the compressor pumps the heat, picked up by the refrigerant in the evaporator, to the higher level in the con-

¹ Moores, C. F., "Refrigeration Helps Save Premature Babies." *Refrigerating Engineering*, Vol. 49, No. 2 (February, 1945), p. 102.

denser so that practical use can be obtained from it. This application is also referred to as reverse-cycle refrigeration, which is somewhat of a misnomer, since the familiar basic refrigeration cycle is still used. However, in a combined heating and cooling system there is a reversing of the

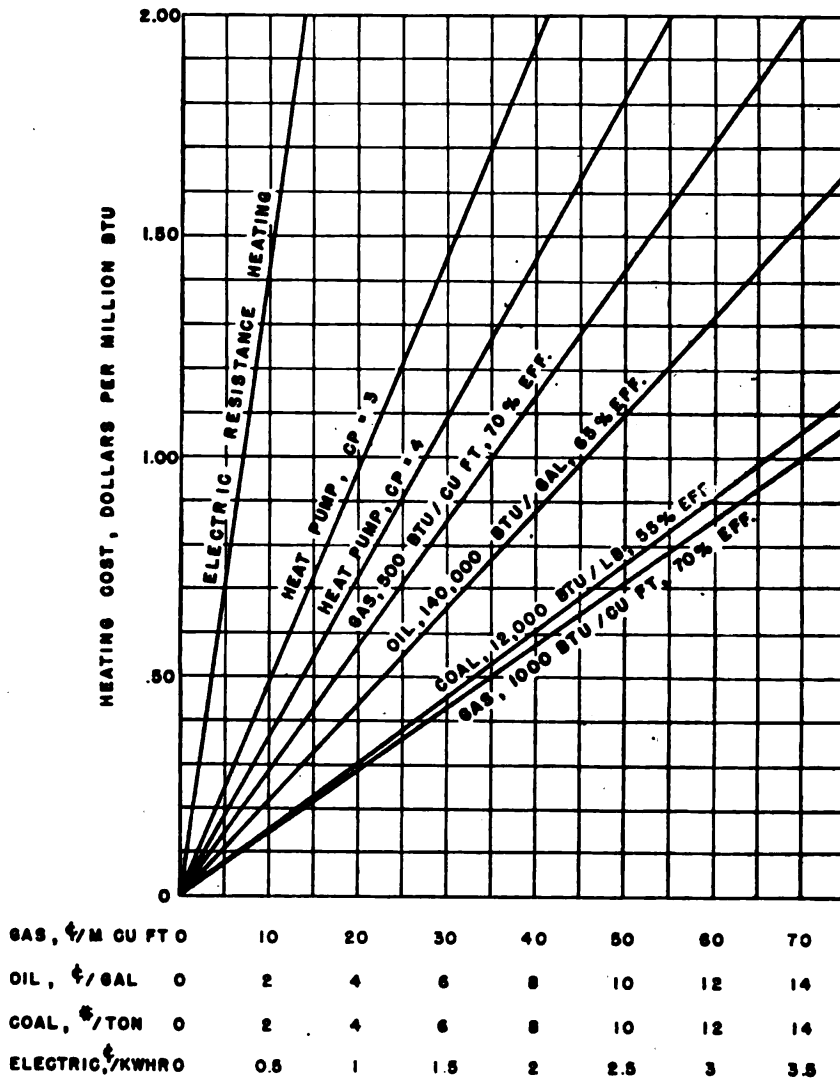


Fig. 17.1. Comparative fuel costs for heating.

utilization of the energy, since the application of the unit is changed by one means or another to give heating or cooling in the space as desired.

This method of heating was first proposed by Lord Kelvin in 1852. Very few installations were made before 1930, since neither the initial nor operating costs could compete with those for conventional systems. On first thought this may not seem reasonable, since the heat available may easily be three or more times the equivalent heat energy supplied to the compressor. This ratio of output over input is called the coefficient of performance for the heat pump and is explained in Chapters 4 and 6. The high operating costs have been due to the cost of energy required to

drive the compressor. But the gradual decrease in electric power rates, combined with the increase in the cost of coal, oil, and gas, particularly since 1940, has changed the economic picture.

Comparative costs of the heat energy required for heating with the heat pump and with the more conventional systems are shown in Fig. 17.1. Note that the relative positions of the curves have no significance but depend on the abscissa scales used. The efficiencies of the conventional systems will vary, and true comparisons depend upon many factors. The efficiencies used were arbitrarily selected in order to have some basis for comparison but should not be far from those obtained in the average forced-air domestic system after a few years of operation. It is suggested that the reader draw in his own line for the fuel and efficiency of interest to him.

Comparative fuel and electric rates taken from Fig. 17.1 that would result in equal costs per million heat units are shown in Table 17.1. This table indicates that, for the heating values and efficiencies assumed, a cost for heat energy of \$1 per million Btu delivered would result from the use of coal costing \$13.20 per ton, or oil costing 9.1 cents per gallon, or electricity costing 1.37 cents per kilowatt-hour and applied to a heat pump with a c.p. of 4.

TABLE 17.1
COMPARATIVE FUEL AND ELECTRIC RATES THAT PRODUCE EQUAL HEAT-ENERGY COSTS

Costs per Million Heat Units	\$0.80	\$1.00	\$1.20
Heat pump—c.p. of 4 at.....	1.10¢ per kwhr	1.37¢	1.64¢
Heat pump—c.p. of 3 at.....	0.82¢ per kwhr	1.02¢	1.23¢
Coal: 12,000 Btu per lb—55% eff at.....	\$10.56 per ton	\$13.20	\$15.84
Oil: 140,000 Btu per gal—65% eff at.....	7.3¢ per gal	9.1¢	10.9¢
Gas: 1000 Btu per cu ft—70% eff at.....	56¢ per 1000 cf	70¢	84¢
Gas: 500 Btu per cu ft—70% eff at.....	28¢ per 1000 cf	35¢	42¢
Electric resistance heating at.....	0.27¢ per kwhr	0.34¢	0.41¢

The heat pump requires the availability in sufficient quantity of a dependable heat source or medium that can supply heat to the evaporator and can be cooled by it. The three basic heat sources that have been used are air, water, and the earth, as discussed in Chapter 6. Figure 17.2 shows a unit made in 3-hp to 10-hp capacities that has been installed in residential and commercial applications in California and in several of the Southern states and uses air as the heat source.

Since World War II the greatest emphasis has been placed on domestic applications. The development of higher speed and hermetically sealed compressor units using nontoxic refrigerants, together with more efficient heat transfer equipment, has made possible the production of compact

self-contained units that should be competitive with other types of equipment used for year-round air conditioning. Present indications are that the initial cost of the heat pump can compete with other equipment only when year-round air conditioning is desired. Some thought has been given to using combinations of air and water or of air and earth as heat sources and also to using heat-storage systems to reduce costs.

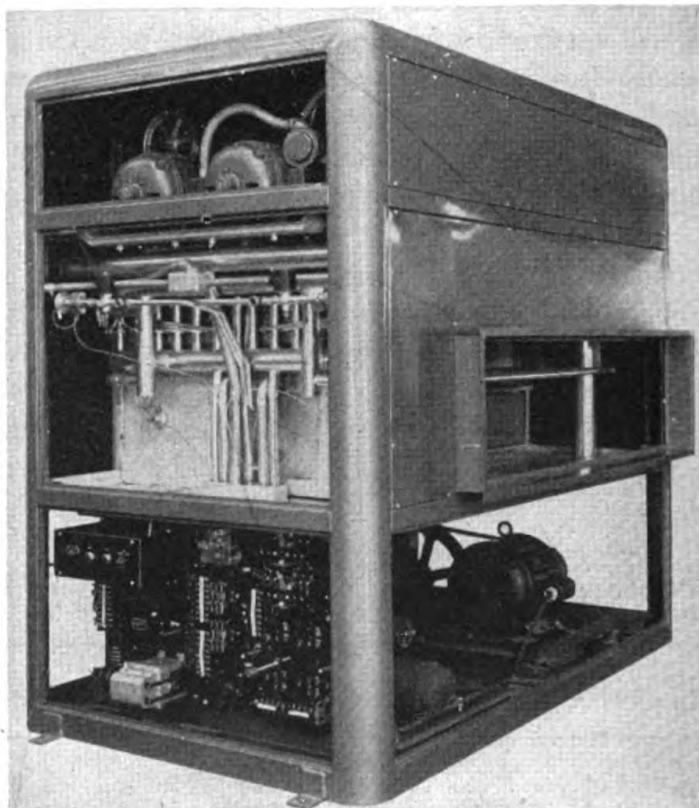


Fig. 17.2. Air-to-air heat-pump unit. Courtesy Drayer-Hanson, Inc., Los Angeles.

During and prior to World War II most installations were for industrial and commercial applications, mainly for electric utility offices but also including theaters, stores, and factories. The critical fuel situation in Switzerland during the war prompted installations there in industrial plants using evaporation or distillation processes. These systems have operated at coefficients of performance higher than those obtained for other installations, as high as 14 in some instances. Consideration might well be given to using the waste condenser heat from a locker-plant refrigerating system to supply part or all of the heat for the office and processing rooms (see §19.5).

Figure 17.3 shows a unit proposed for use with a ground coil.

Words of caution should be heeded by those who intend to design a heat-pump system. A fan capacity and supply-air ducts larger than those for a conventional forced-air heating system are needed to handle

the greater air quantities required because of the lower supply-air temperature. Economy of operation indicates using air temperatures of 100 to 110 F instead of 140 F or higher. If the condenser and evaporator are

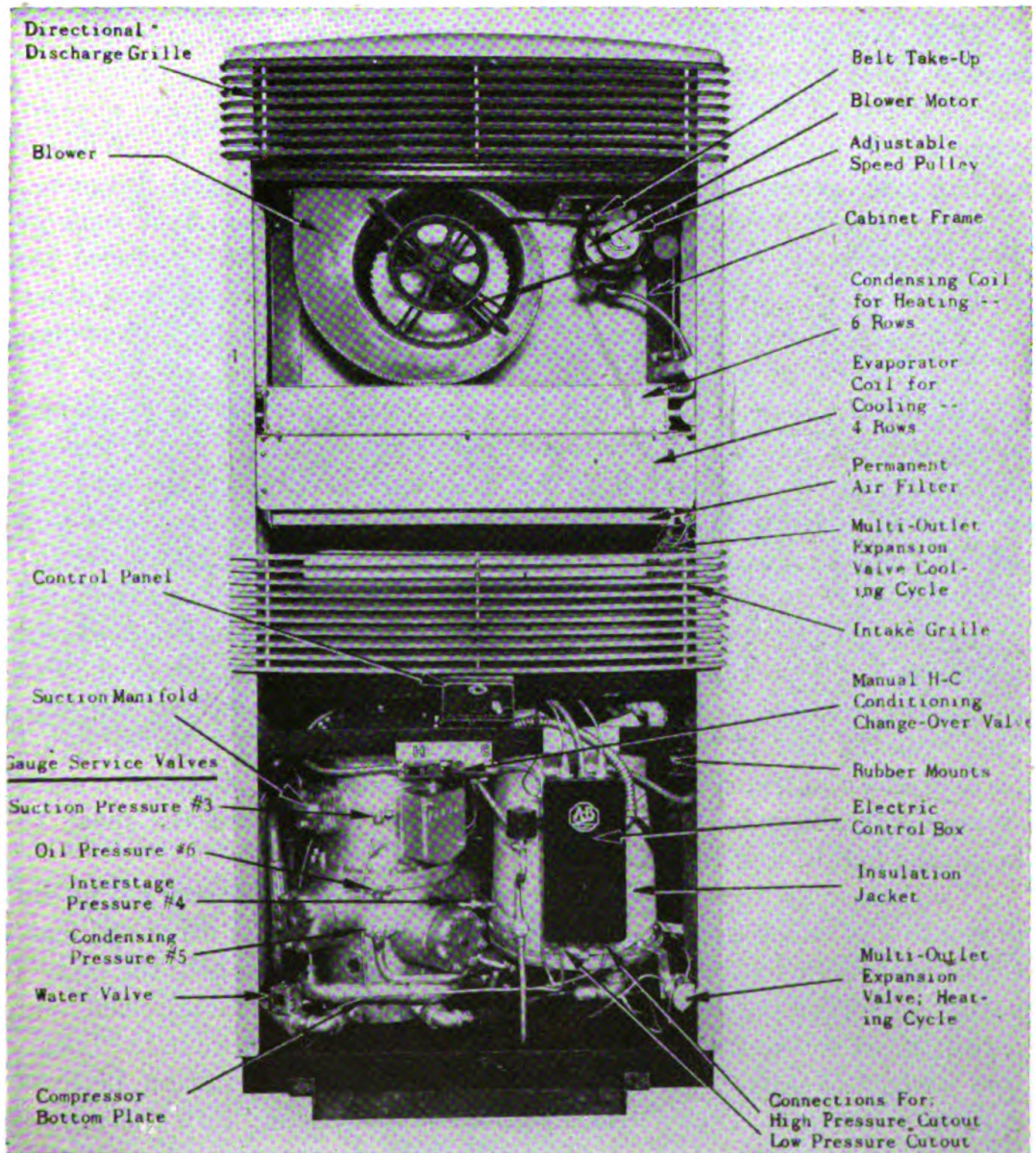


Fig. 17.3. Ground-to-air heat-pump unit. Courtesy General Engineering and Mfg. Co.

to interchange functions as the system switches from heating to cooling, they must each be designed to function satisfactorily as either unit for the loads involved. One manufacturer of self-contained units decided to use two separate circuits, an arrangement that involves two evaporators and two condensers, and to switch from one to the other, even though more equipment is required. When air is used as the heat source the ducts can be arranged so that dampers reroute the air flow as desired.

If refrigerant coils are placed in the ground, the length of circuits and problems of pressure drop and oil return must be considered, as well as the more important question of the adequacy of the heat source year after year.

17.3. Medical and Surgical Aids. Members of the medical profession have worked with members of the refrigerating engineering profession, have carried on original experiments in refrigeration, and have applied refrigeration in their own work for many years. In fact, Dr. William Cullen, who graduated from the College of Physicians and Surgeons, Glasgow, in 1729, did some important preliminary development work in refrigeration. In 1755 he wrote in his "Essay on Cold Produced by Evaporating Fluids" how he froze water by exhausting nitrous ether from a vessel placed in the water. In 1781 Tiberius Cavallo, a physician born in Naples in 1749 and who later went to England, wrote in the *Philosophical Transactions* of London about his experiments on cooling by evaporation of various ethers and by using air. Dr. John Gorrie, a physician born in Charleston, S. C., developed an ice-making machine before 1844. He used ice to condition air for his yellow-fever patients and by 1851 demonstrated the value of ice packs and cooled rooms in the treatment of certain tropical fevers. His patent, No. 8080, was ridiculed by the American press but was given favorable publicity in England and served as a model from which inventors the world over patterned their work. In 1869 Dr. Henry Peyton Howard, a physician of San Antonio, Tex., was the first to transport refrigerated beef by ship. He equipped his own ship and transported the beef from Palacios, Tex., to his own cold-storage warehouse in New Orleans and then distributed the meat to hospitals, hotels, and restaurants.²

Two major applications of refrigeration in modern surgery are in the prevention of shock and as an anesthetic in operations on arms and legs. Controlled cooling applied gradually from the outside can help the body's natural reactions against ordinary shock by reducing metabolism and permitting a concentration of oxygen and plasma at the wound. Localized refrigeration has been used as an anesthetic for amputation, because chilling is the only known treatment for anesthetizing both the nerves and the entire protoplasm, and it apparently prevents excessive blood-clot formation. Burns also benefit from moderate cooling for a number of days or weeks.³ Freezing of local parts of the body can be endured for brief periods, and isolated cell structures can be kept frozen in a refrigerator for days, be replaced, and resume functioning. But in general, temperatures below 40 F are not used.

² Woolrich, W. R., "Mechanical Refrigeration—Its American Birthright." *Refrigerating Engineering*, Vol. 53, No. 3 (March, 1947), pp. 196–199.

³ Potter, R. H., "Refrigeration Has Become an Instrument for Healing." *Refrigerating Engineering*, Vol. 52, No. 4 (October, 1946), pp. 306–309.

Results of chilling experiments by doctors over the past several years have been reported in the literature.⁴ Crossman and Allen state that it might even be possible to restore a cleanly amputated limb or finger if it is quickly packed in ice and sent to the hospital with the patient.

Ice is generally used, but specially designed direct expansion and brine-chilled blankets as well as refrigerated air chambers have been made that are used to lower skin temperatures down to 40 F. Wider

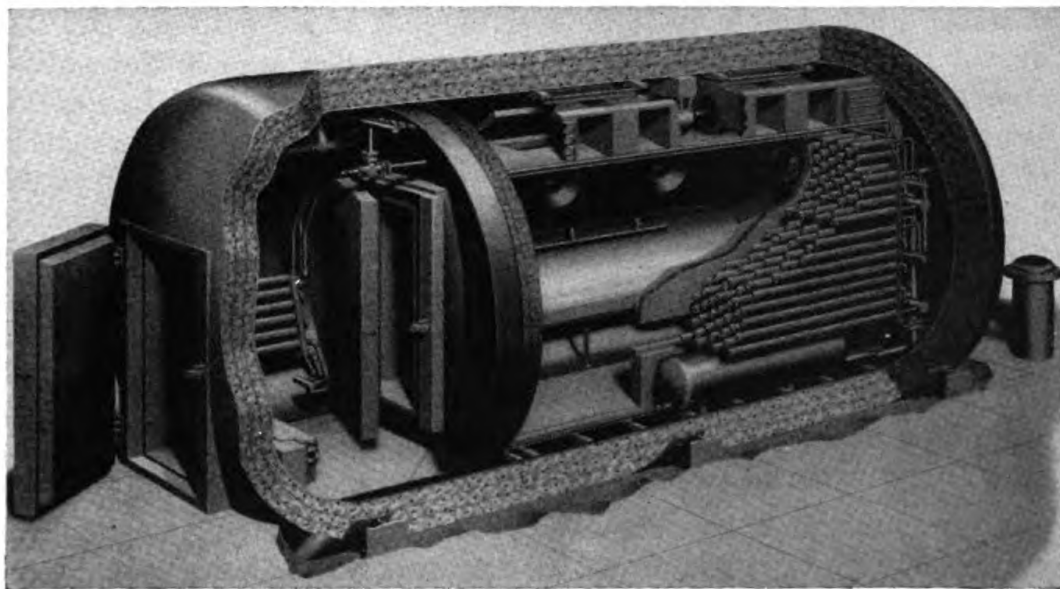


Fig. 17.4. Stratochamber. Courtesy York Corp., York, Penna.

applications are continually being found, and research on new and better types of equipment is needed.

17.4. Creation of Artificial Atmospheres. This term has reference to applications other than ordinary comfort or industrial air conditioning.

Stratochambers. Since man desires to travel in the stratosphere, it is necessary that vital parts of the machine to be flown, the instruments to be used, all necessary equipment, and even the fliers themselves be subjected to atmospheric conditions like those to be encountered, and their reactions studied. It is more practical to simulate the stratospheric conditions in a stratochamber down on earth and place the items to be observed in it than to do much experimenting aloft.

Various types and sizes of chambers have been built, but most of them are divided into a lock, or vestibule, and a main chamber similar to Fig. 17.4. In chambers to be used to study human reactions and certain equipment, a "climb" chamber is built within the main chamber. Refrigeration equipment is needed that can reduce the temperature in

⁴ Crossman, L. W. and Allen, F. M., "Surgical Refrigeration and Preservation of Tissue." *Journal of the American Medical Association*, Vol. 133, No. 6 (Feb. 8, 1947), p. 377.

the climb chamber from 70 F to -70 F in less than 12 min, and vacuum equipment is needed to reduce the pressure from that at sea level to that at 45,000 ft or higher altitude in less than 5 min. These two changes must take place simultaneously to simulate climbing. In certain equipment-testing chambers, snow, ice, and frost forming devices must also be employed.

A chamber designed for physiological tests on six men in electrically heated suits would be supplied with 10 cfm of ventilation air. The climb chamber would be about 18 ft long, the lock about 4 ft long, and the diameter 14 ft. The total load for such a chamber at -70 F would be about 2.5 tons. In determining the load a transmission factor of 0.1 Btu(in) per (hr)(sq ft)(deg F) is recommended to account for the release of sensible heat from the mass of material even though the theoretical U is 0.025. The occupant heat is taken as 1380 Btu per hour per person. The ventilation air should be dried by an adsorbent so that only its sensible heat is included in the chamber cooling load.

Using a two-stage Freon-12 system at -85 F suction temperature for 2.5 tons with interstage liquid and gas cooling would require a 30-hp low-stage and a 20-hp high-stage machine. A flooded evaporator with a surge drum and liquid refrigerant pump is recommended. The refrigerant cycle would then be to have the refrigerant vapor drawn from the surge drum into the low-stage compressor, discharged through a double-pipe gas cooler, drawn into the high-stage compressor, and discharged into the water-cooled condenser. The liquid would flow through a float-valve control to a flash-type liquid cooler, to a double-pipe oil still, and then to the surge drum through a second float-valve control. The pump circulates the liquid from the surge drum through the evaporator and back to the surge drum. More details are given in the *Refrigerating Data Book*.⁵

Wind Tunnels. The largest refrigeration plant in the United States at the time of installation in 1944 is at the National Advisory Committee for Aeronautics Altitude Test Tunnel in Cleveland, Ohio. Fourteen four-stage Freon-12 centrifugal compressors, each driven by a 1500-hp motor, 260 plate-fin coils cooling 10,500,000 cfm of air, and a 6-ft-diameter flash cooler 55 ft long are used to produce 7500 tons of refrigeration using 58,000 lb of Freon-12 in 8-in. liquid lines and in one 45-in. and four 32-in.-diameter suction lines. The coils are operated flooded, with a Freon pump forcing liquid through them from (and back to) the flash cooler. Conditions comparable to a 50,000-ft altitude can be produced in the 20-ft-diameter test section with air speeds over 500 mph. Standard tests can be conducted at controlled temperatures from 59 to -67 F, and it is

⁵ *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed. New York: American Society of Refrigerating Engineers, 1946, "Stratochambers," Chapter 45, p. 421.

possible to obtain -80°F . In addition to the 21,000 hp for the Freon compressors, considerable power is also required for the pumps, fans, and auxiliary equipment.⁶

There are at least 10 other wind tunnels of various types and sizes in the United States, which provide air velocities up to and in the supersonic region and temperature conditions from 140°F down to -354°F and which use centrifugal refrigeration compressors.

Artificial Snow. In November, 1946, V. J. Schaefer of the General Electric Research Laboratories produced snow from a three-mile stratus and cumulus cloud near the Massachusetts-New York state border by dropping frozen carbon dioxide (dry-ice) pellets from an airplane flying through the cloud. Near-by clouds not touched disappeared without giving up any moisture. Another demonstration was made near Portland, Ore., in April, 1947, and rain has been produced on several occasions since that time. Previously, artificial snow had been produced by similar methods in the laboratory.

The principle involved is that, if clouds contain supercooled water droplets, as they often do, the dry-ice pellets furnish nuclei for the development of raindrops or snow crystals. A pellet the size of a pea is capable of producing tons of snow as it drops 2000 ft through a cloud. A cloud two miles thick can produce about 0.14 in. of rain, but probably only a small percentage of this would reach the ground. The Army, Navy, and Air Force have been continuing to experiment along these lines,⁷ but certain conservative meteorologists are still quite pessimistic.⁸ If practical, possible applications would be the dissipation of winter fogs over airports or the steering of snowfalls away from urban areas and toward winter resorts or toward storage reservoirs for irrigation and water-power uses. This may be a somewhat unusual application of refrigeration, but it illustrates the diversity of the field and may stir the imagination to think of additional new applications.

17.5. Manufacture and Treatment of Metals.

Blast-Furnace Operation. The control of the moisture content in the air supplied to blast furnaces for the manufacture of pig iron was suggested by the first president of the Iron and Steel Institute of England in his address to the membership over a hundred years ago. Conclusive demonstrations of the value of a dry blast in increasing iron production and decreasing fuel consumption were made in actual operation of various plants in the early years of the twentieth century. However, the equipment used at that time proved to be uneconomical.

⁶ Wilson, M. J., "Low-Temperature Refrigeration System." *Refrigerating Engineering*, Vol. 48, No. 5 (November, 1944), p. 369.

⁷ "Weather under Control," *Fortune*, Vol. XXXVII, No. 2 (February, 1948), p. 107.

⁸ Drake, Lawrence, "Rainmakers Are All Wet." *Magazine '48* (March, 1948), p. 61.

Modern centrifugal refrigeration machines and other improved equipment have changed the situation to the extent that one company in Alabama installed a second and then a third unit after realizing the benefits of the first installations. A few installations have been made in Northern plants, but there still seems to be some question of their practicability in the minds of a great number of operators.

No one seems to know what the optimum moisture content is for the air blast. In this application the air humidity is indicated by grains of moisture per cubic foot of air. There is not agreement as to whether 1, 2, or 3 grains per cubic foot is best, but there is no question about a constant air humidity producing pig iron of a more uniform quality.

Atmospheric humidity in the Great Lakes region varies from $\frac{1}{2}$ grain per cubic foot in the winter to 9 grains per cubic foot in the summer. A typical 1000-ton blast furnace is charged every 24 hr with the following:

Ore.....	1775 tons
Coke.....	850 tons
Limestone.....	385 tons
Total	3010 tons

The air required is 4390 tons, or about 59 per cent of the total charge.^{9,10} This is roughly 80,000 cfm so that a 6-grain reduction would require

$$\frac{80,000 \text{ cfm} \times 6 \text{ gr/cf} \times 60 \text{ min/hr} \times 1060 \text{ Btu/lb}}{7,000 \text{ gr/lb} \times 12,000 \text{ Btu/(hr)(ton)}} = 364 \text{ tons of refrigeration}$$

Heat-Treatment. Since the conventional treatment of steels failed to produce a uniform structure in special steels, processes involving low temperatures, in some cases as low as -130°F , were developed. Liquefied gases and dry ice have been used, but with modern equipment mechanical refrigeration can also be used. A temperature below -70°F is hard to maintain with dry ice.

Low-temperature treatment was applied to stainless steel back in 1914 to develop spring properties. Research work has been carried on continuously, so that entirely satisfactory processes have now been developed for low-temperature treatment of high-speed tool steels and for other steels of various contents of carbon, chrome, molybdenum, nickel, and tungsten.¹¹

Aluminum-alloy rivets tend to harden with age after heat-treatment.

⁹ Dunne, R. V.D., "Dry Blast and Production for War." *Refrigerating Engineering*, Vol. 44, No. 1 (July, 1942), p. 19.

¹⁰ Dunne, R. V.D., "Economic Aspects of Pre-Compression Refrigerated Dry Blast." *Iron and Steel Engineer*, Vol. 19 (July, 1942), p. 56.

¹¹ *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed. New York: American Society of Refrigerating Engineers, 1946. "Low Temperature as Applied to Metals," Chapter 43, p. 409.

This hardening increases their strength but decreases their workability. Special alloys and heat treatments were developed to overcome this tendency including alloy A17S, which can be worked after aging unless the forming operation is severe. Alloys 17S and 24S must be chilled promptly after quenching. The lower the storage temperature, the longer the hardening is retarded. At 0 F no hardening occurs for at least a week or longer. In some cases -30 to -45 F temperatures are used.¹² Special low-temperature containers are used to transport the alloys from the heat-treating room to the place of fabrication.

Miscellaneous Applications. Cooling of metal parts to obtain expansion fits instead of heating the other parts to obtain shrink fits is now practiced to eliminate the problems of oxidation and the need for final finishing. Temperatures required vary from -50 to -90 F. Portable low-temperature chambers like those for aluminum alloys may be used.

Refrigeration is needed to cool the electrodes in modern welding equipment. A cooled antifreeze circulating system has been found to be most practical.¹³ Other applications include cooling brine baths used for quenching (fluctuations in metal hardness result from variations in bath temperatures), cooling electroplating baths, cooling anodizing solutions, and cooling gear-cutter and lubrication oils.

17.6. Oil Refining and Synthetic-Rubber Manufacturing. These two applications are related, since in 1943 the announced goal of synthetic-rubber production during World War II was to be about 58 per cent buna S and 12 per cent butyl rubber, and the constituents of both of these are made mostly from petroleum hydrocarbons.¹⁴

Oil Refining. The petroleum industry was one of the first to use refrigeration in great quantities and continues to require large installations. In 1885 partial dewaxing of lubricating oils was done by "cold-settling" in tanks. Because compression machines in the early days suffered from lubrication and other difficulties, absorption units were used more extensively, since they gave equally satisfactory results with less trouble. The new large, improved absorption units described in Chapter 15 were developed largely from refinery experiences, and the majority of the new installations are in the oil industry (see Fig. 17.5).

Refrigeration is needed off and on throughout the refining process, from the time the crude oil enters until the finished products leave. One main purpose of refrigeration is to chill the oil at various stages in the process until the wax crystallizes and can be separated out. A temper-

¹² Keller, F., "Refrigeration of Aluminum Alloys," *Refrigerating Engineering Application Data* 35. *Refrigerating Engineering*, Vol. 49, No. 2 (February, 1945), p. 148.

¹³ *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed. New York: American Society of Refrigerating Engineers, 1946. "Cooling of Electrodes in Welding," Chapter 54, p. 507.

¹⁴ *Ibid.*, "Oil Refining," Chapter 52, p. 489.

ature of -50°F or lower is needed to produce oils having a pour point of 0°F . Various procedures are used as described in the reference of footnote 14. Refrigeration is also used in the carefully controlled low-temperature recovery in the fractional distillation of the lighter hydrocarbons and to

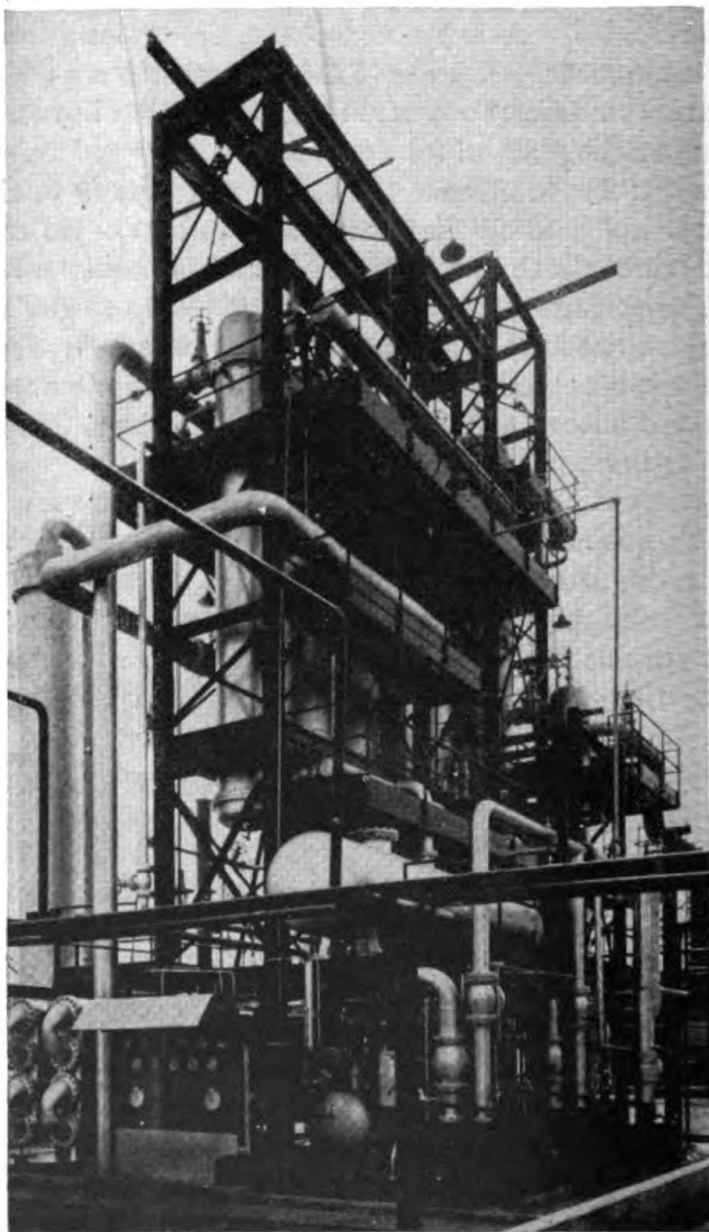


Fig. 17.5. 700-ton absorption unit for an oil refinery.
Courtesy Worthington Pump and Machinery Corp.

remove the heat of absorption when sulfuric acid is added in the sulfur-removing processes.

In general, these applications may be classified as influencing solubility relationships and securing a separation of components, reducing vapor pressure of volatile components, and securing selective chemical reactions.

In an absorption plant, refrigeration requirements range from 10 to 30 tons per million cubic feet processed daily.¹⁵ Steam-ejector refrigeration is sometimes used, as shown in Fig. 17.6.

Synthetic Rubber. Literally, this expression is a misnomer, it should be "rubber substitute" or "synthetic rubberlike material," since these

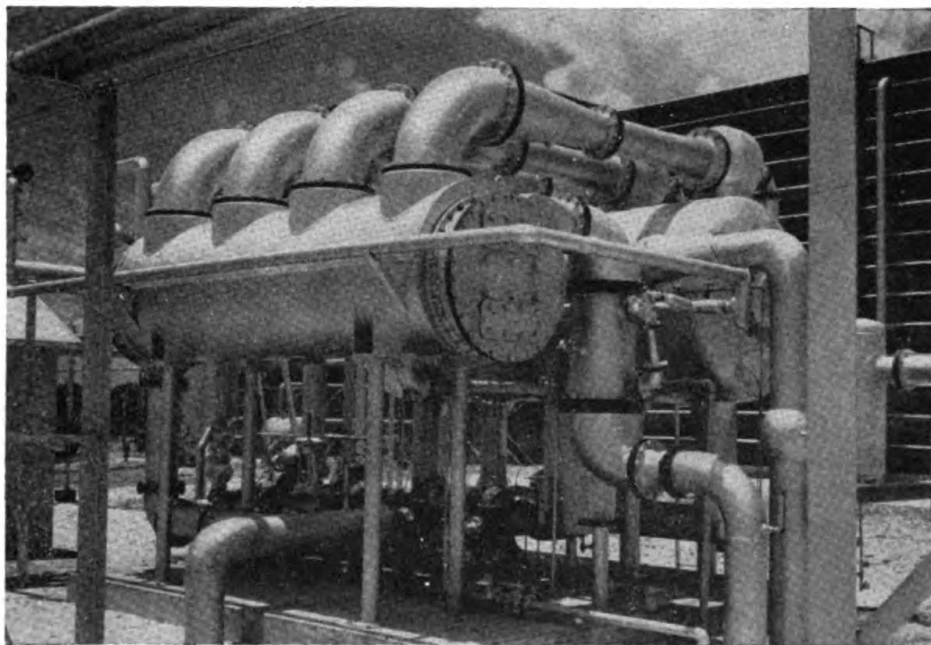


Fig. 17.6. 250-ton steam-jet unit for an oil refinery. Courtesy Worthington Pump and Machinery Corp.

new products are not chemically the same as rubber. But the expression is like many others in that its definition must be based on common usage.

The wartime expansion in this field was tremendous, and work was concentrated on the production of four types of materials, buna S, butyl, thickal, and neoprene. Incidentally, neoprene had been used for several years previous to the war in the shaft seals for refrigerating machines. Production details were not publicized during the war, and research developments have been changing procedures, but certain general information has been made available.¹⁶

Buna S, which was in most demand, is by weight, 1 part styrene and 3 parts butadiene. Styrene may be made from refinery gas, and butadiene may be made from alcohol, petroleum products, or other materials. Butyl is made from refinery gases.

Butadiene, at atmospheric pressure, boils at 23.5 F, and storage at

¹⁶ Cole, Otis D., "Some Interesting Properties of Synthetic Rubber." *Refrigerating Engineering*, Vol. 45, No. 6 (June, 1943), p. 405.

¹⁵ Cannon, C. W., "Refrigeration Applications in Petroleum Processing." *Refrigerating Engineering*, Vol. 51, No. 3 (February, 1946), p. 130.

32 F and 3 psi pressure is recommended. Some producers also assert that styrene should be stored at 32 F to maintain its purity even though its boiling point is 295 F at 14.7 psia.¹⁷ In addition, there are extensive refrigeration requirements for the recovery of butadiene (one plant using 3000 tons of refrigeration at -5 F to obtain 55,000 tons of butadiene per year) and for some of the subsequent processing methods in manufacturing synthetic rubber.

17.7. Chemical and Related Industries. Refrigeration is a necessary part of many chemical processes. One striking example was tragically demonstrated by the explosion in Los Angeles, Feb. 12, 1947, which killed 15 persons, injured more than 150, and damaged about 300 near-by buildings with a loss of nearly \$2,000,000. Investigation disclosed that an electroplating company had been experimenting with a solution of perchloric acid and acetic anhydride for electrolytic brightening of aluminum. This mixture explodes at temperatures above 80 F, so that constant refrigeration is required. One survivor disclosed that employees had been told of the cooling-system failure and were dashing out when the acid tank, containing 350 gal, exploded. No traces of the bodies of the chemist or his assistant could be found.¹⁸ Of course such a mixture should not be used, at least in such a quantity or in a locality where people and property may be exposed to the possible danger in case of any failure.

In general the chemical engineer requires cooling for processes involving the following: control of reaction rates, control of solubilities, liquefaction of gases and vapor, and solidification of liquids. For economic reasons water is usually used if the available supply is satisfactory. Hence geographical location is important for many plants. When water cannot do the complete job, refrigeration is used either as a supplement or to do all the cooling. The form of refrigeration used depends upon the product and process involved. In some cases water ice or dry ice is mixed with the product, in other mechanical or absorption refrigeration is used, or the material being processed may be used to cool itself. One of the largest installations of refrigeration equipment in the world is required for the ethyl-alcohol plant of the Shell Chemical Company in Houston, Tex., and 24,000 hp of refrigeration equipment was installed for the atomic-pile plant in Hanford, Wash.

The heat evolved in the fermentation of sugars to alcohol must be removed, since a rise in temperature affects the efficiency of the operation. Cooling is needed to control the reactions in making chlorine bleaching materials. The common reactions of cellulose may easily proceed too far unless controlled carefully by refrigeration. Cellulose is the raw

¹⁷ Roe, Arthur, "Refrigeration in the Oil Refining Industry." *Refrigerating Engineering*, Vol. 46, No. 4 (October, 1943), p. 225.

¹⁸ "Refrigeration Failure." *Heating and Ventilating*, Vol. 44, No. 3 (March, 1947), p. 102.

material for a type of rayon, cellophane, nitrocellulose, and many plastics and lacquers. Minor variations in the process may change the product from the base for a fabric, plastic, or lacquer to an explosive.

Purification of natural gas and of some manufactured gases is accomplished by cooling the gas enough to liquefy the impurities. Combined compression and cooling are used to liquefy chlorine and other gases for ease of handling and shipment. The distillation and recovery process of liquids with low boiling temperatures require refrigeration when condensing temperatures below those possible to obtain with water are needed. These include carbon disulfide and tetrachloride, ethyl and methyl chloride, ether, and others.

Refrigeration is needed for direct chilling and for the air conditioning required in the manufacture of gelatin and glue. Cottonseed oil is stored at 14 to 40 F to allow the stearin to collect and settle out. The clear oil can be filtered off and then will not become cloudy. The stearin is used to make vegetable lard by a compounding process that also requires refrigeration. The fats from alcoholic solutions of perfume concretes are separated by freezing so that the special odors will not be destroyed by high-temperature methods.

Many salts are crystallized by cooling and by evacuation of the vapor over the solution. Refrigeration is used to control accurately the percentage of water of crystallization in the final product.¹⁹

One of the most important reactions that requires close temperature control to prevent loss of product and possible hazards is nitration. Aniline dyes and pharmaceuticals are made from nitrobenzene, and 865 Btu are evolved and must be removed for each pound of nitrobenzene made by nitration of benzene. The explosives industry finds refrigeration essential, since most of the explosives consist of nitrated materials.²⁰

17.8. Ice Making. Although the ice wagon is a thing of the past, over 54,000,000 tons of ice was sold in 1946 and delivered mainly by truck. This figure represents some 86,000,000 ton-days of refrigeration actually produced. (A ton-day of refrigeration is 288,000 Btu, or a ton of refrigeration operating for a day—approximately the latent heat of 2000 lb of ice.) Most of us are so accustomed to using and seeing mechanical refrigeration equipment that we do not realize there still is such a demand for delivered ice. Recent technological development in refrigeration has been concerned mainly with equipment to replace ice, so that modern ice-making plants differ little from some of those proposed before 1890.²¹ The latter, however, were an improvement over the plants used to produce

¹⁹ *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed. New York: American Society of Refrigerating Engineers, 1946. "Chemical Industries and Refrigerant Manufacture," Chapter 50, p. 472.

²⁰ *Ibid.*, "Powder, Explosives and Pyrotechnics," Chapter 51, p. 483.

²¹ *Ibid.*, "Ice Making Plants," Chapter 46, p. 429.

artificial ice during and just after the Civil War. United States census reports indicate there were at least four plants making ice by 1870. One method was to freeze distilled water in cans, condensing it from steam-using equipment; also, river, pond, or well water was frozen on two sides of refrigerated vertical plates 12 ft high by 20 ft long. Plants using these methods have been replaced by "raw-water can plants" somewhat similar to those designed over 60 years ago.

In order to produce clear ice, air agitation is used to force the dissolved salts and coloring matter to the center of the can as clear ice crystals form and grow inward from the can surface. When only 3 to 4 gal of concentrated salt solutions in the core remain unfrozen, they are pumped out and replaced with cooled fresh water.

Cans are made in various shapes of 25 to 400-lb capacities. The 300-lb can is one of the most popular; the smaller sizes are used mainly in restaurants and institutions that have their own ice-making plants. The fact that the actual weight of ice in a 300-lb can is about 320 lb must be considered when designing the equipment. The plant-capacity rating, however, would be based on 300 lb per can.

In most plants brine is chilled to about 12 F by ammonia evaporating at 3 to 7 F. Raw water cannot freeze without cracking at temperatures below 10 F, whereas the limit for distilled water is 6 F. An 11 × 22-in. can freezing 320 lb with 12 F brine will freeze 280 lb in 24 hr but will take 14 hr to freeze the remaining 40 lb. The refrigeration equipment required to produce 1 ton of ice per 24 hr will depend upon the filling-water temperature, brine temperature, construction details, and other factors. Normally, for 70 F entering water and 12 F brine practically 1.6 tons of refrigeration is required per ton of ice made.

Since World War II there have been unprecedented demands for the unitary type of ice-making equipment, particularly for flake-ice machines such as shown in Fig. 17.7 and for automatic ice-cube machines used by restaurants, bars, hotels, hospitals, and institutions.

Skating Rinks. Artificial rinks freeze a sheet of ice by direct expansion cooling or by a brine circulating system. The design of the floor and piping system requires careful study, and there is a definite type of floor construction that is best for each particular installation.²² The simplest plan has the pipe coils supported in sand placed directly on the ground. Such a floor requires considerable maintenance, and the pipes soon corrode. The more complex constructions have concrete bases with insulation and have the pipe coils in concrete. The insulation is to aid the change from an iced floor at 15 or 20 F to a dry 70 F floor, or vice versa, in a few hours as desired in the modern arenas.

²² *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed. New York: American Society of Refrigerating Engineers, 1946. "Skating Rinks," Chapter 47, p. 445.

Steel pipe of 1-in. or 1½-in. size is generally used. Wrought iron may last longer, but it is more expensive. Experience indicates most satisfactory temperature distribution and heat absorption is attained by embedding 1-in. pipe in concrete on 4-in. centers. The brine-temperature rise should not exceed 3 F, and usually a grid circuiting system gives best results.* Air-relief openings should be included in the piping system. Recently some small ice-show rinks in hotels have used continuous refrigerated plates.

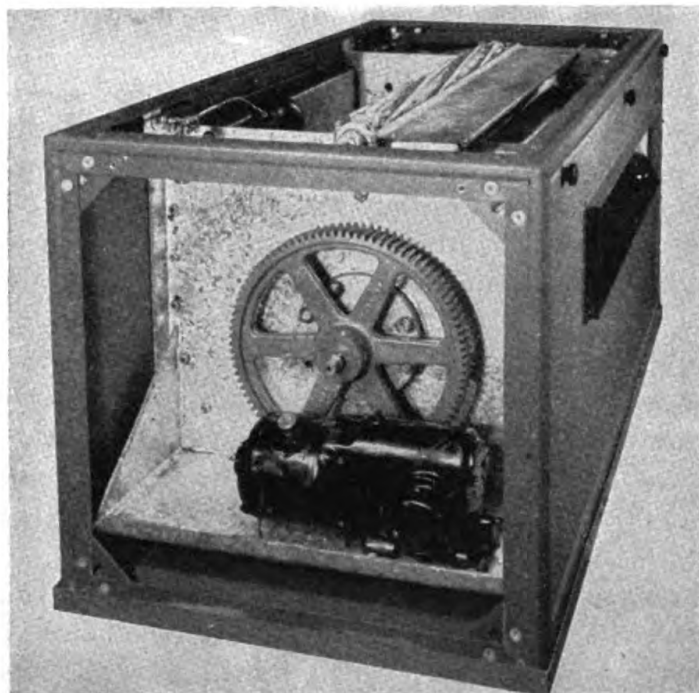


Fig. 17.7. Flakice machine. Courtesy York Corp., York, Penna.

A dry ice at 32 F is best for general skating. Colder ice cracks, becomes rough, and gets covered with snow. Thus the brine temperature must be controlled to close limits. An electric resistance temperature-indicating system is recommended since the heat gain to the ice may change rather quickly and a 0.1 F variation in ice temperature is important. Since the total refrigeration load may be three times as great at one time as at another, step control of compressor capacity should be provided. Changing the air temperature over the ice is a good quick method of ice-temperature control. Refrigeration capacity varies from 0.5 to 0.85 ton per 100 sq ft of floor. Many construction and operating details may be found in the references of footnote 22.

17.9. Miscellaneous Applications.

Extreme Low Temperatures. Various actual and possible applications in the cryogenic field were mentioned in Chapter 16, and several articles on the subject were listed in that chapter's bibliography reference 3.

Included in the results of research during World War II is the development of a new method for producing liquid hydrogen in amounts of 3 to 5 liters to use where portability and simplicity are important. A small cryostat also was designed, in which a liter of hydrogen slush can maintain temperatures from 14 to 20 K for 20 hr.²³ This combined apparatus makes possible the use of certain low-temperature bolometers, particularly for infrared measurements and for spectroscopic work.²⁴ A bolometer is an instrument for measuring small quantities of radiant heat.

Superconducting bolometers make use of the phenomenon that certain materials lose most of their resistance to the flow of an electric current when subjected to temperatures between absolute zero and about 15 K. The transition from an appreciable electric resistance to very little resistance takes place within a very small temperature change. In order for the bolometer to be sensitive to slight variations in infrared radiation that would change its temperature slightly, and hence its electric resistance, it must be closely maintained at a constant temperature in the transition range.

An early bolometer used tantalum cooled by a helium bath boiling at 3.22 K under reduced pressure. Further research disclosed that the resistance transition for columbium nitride occurred near 15 K. This temperature can be obtained with liquid hydrogen and reduce the refrigeration cost to about one-tenth of that necessary to operate in the helium range.²⁵

Dr. Andrews and his co-workers at Johns Hopkins University accidentally discovered that the columbium nitride bolometer at about 14 K (−435 F) serves as a radio receiver by itself. Several stations could be heard through the loudspeaker which was previously used to indicate the presence of objects “seen” by the bolometer. Practical applications may be found in transoceanic and other long-distance radio, but the cost of the refrigeration equipment limits its general practicability in ordinary radio work.

The development of a helium cryostat by Professor S. C. Collins of Massachusetts Institute of Technology to provide temperatures below 2 K (−457 F) has been announced.²⁶ Gaseous helium is circulated through a heat exchanger at 200 psi, then expands through an engine

²³ DeSorbo, W., Milton, R. M., and Andrews, D. H., “New Cryogenic Methods of Using Liquid Hydrogen.” *Chemical Reviews*, Vol. 39, No. 3 (December, 1946), p. 403.

²⁴ Milton, R. M., “A Superconducting Bolometer for Infrared Measurements.” *Ibid.*, p. 419.

²⁵ Andrews, D. H., Milton, R. M., and DeSorbo, W., “A Fast Superconducting Bolometer.” *Journal of the Optical Society of America*, Vol. 36, No. 9 (September, 1946), p. 518.

²⁶ “Low-Temperature Research.” *Heating and Ventilating*, Vol. 43, No. 12 (December, 1946), p. 93.

where it does work. The exhaust from the engine passes to the heat exchanger to cool the incoming helium. The entire unit weighs about $\frac{1}{2}$ ton and will find application in metallurgical research on thermal conductivity, embrittlement, and other characteristics.

Lyophilization. A comparatively new application of refrigeration is in lyophilization, or desiccation in the frozen state. This process is used for drying substances that change chemically or physically when heated. It is used principally in the pharmaceutical field but may be applied elsewhere. The product to be dried, for example blood serum or penicillin, is first frozen and then placed in a chamber that can be evacuated. Refrigeration is used not only to freeze the substance but also to remove the moisture ahead of the vacuum pump in the air being evacuated from the product storage chamber. The vacuum pump aids drying by creating and maintaining an extremely low vapor pressure in the chamber.

Plumbing. In 1947 a British firm announced a portable unit designed to stop the flow of water quickly in a leaking pipe to permit repairs. Refrigerant is circulated through a tool that freezes the water in the pipe at the point of application and is useful if a handy shutoff valve has not been provided.²⁷

Building Construction. Civil engineering projects often require or are greatly aided by refrigeration. Methods of controlling moisture-laden soil by freezing have been used for many years to prevent cave-ins and to obtain soil samples. Refrigeration has reduced the time required to sink shafts and make tunnels through water-bearing strata, also simplifying the work. Other applications include the cooling of many massive concrete structures during construction. Cooling is used to remove the heat of hydration and to hasten the shrinkage of concrete. It is particularly useful in the construction of dams, which are ordinarily made in 5-ft lifts. The time between the pouring of the lifts can be reduced, and the lifts are more nearly in their final shape before being loaded and placed under stress.

Ice and chilled water have been used in making the concrete mix, and copper or thin steel coils, usually of 1 in. O.D. and spaced from $2\frac{1}{2}$ to 6 ft on centers, have been embedded in the structure. Chilled water is then circulated through the coils.²⁸

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²⁷ *Air Conditioning and Refrigeration News*, Vol. 50, No. 15 (April 14, 1947), p. 21.

²⁸ *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed., 1946. New York: American Society of Refrigerating Engineers, "Refrigeration in Engineering Construction," Chapter 49, p. 462.

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PROBLEMS

17.1. Determine the cost of obtaining 1,500,000 Btu (a) with 12,000 Btu per pound of coal at \$15 per ton, (b) with 140,000 Btu per gallon of oil at 9.6¢ per gallon, (c) 1000 Btu per cubic foot of gas at 75¢ per 1000 cu ft, and (d) for a heat pump with a coefficient of 4 with electricity at 1.5¢ per kilowatt-hour.

17.2. Using the data in Fig. 12.5 and assuming that the compressor capacity plus the horsepower input is available as heat in the condenser, calculate the coefficient of performance for a heat pump at 105 F condensing and 25 F suction. Assume that the actual energy input to the compressor motor is 12.5 per cent greater than the horsepower input to account for the motor efficiency.

17.3. Determine the coefficient of performance of a heat pump at 40 F suction and at (a) 100 psi, (b) 120 psi, and (c) 140 psi condensing pressures, using Fig. 12.5. Assume that the compressor capacity plus the horsepower input are available as heat at the condenser and the actual compressor motor input is 12.5 per cent greater than the horsepower input.

17.4. How many ton-days of refrigeration are required to produce 10 tons of ice per day at 12 F from water at 72 F if miscellaneous losses are 15 per cent of the chilling and freezing load?

17.5. Calculate the tons of refrigeration required to freeze in 4 hr $\frac{1}{8}$ in. of 50 F water on a skating rink 200×100 ft if the ground temperature is 50 F and the air temperature above the rink is 60 F. The heat-transfer coefficient from the air to the 32 F ice is 1.2 Btu per (hr)(sq ft)(F), and the heat-transfer coefficient from the ground to the 20 F ice 0.06 Btu per (hr)(sq ft)(F).

CHAPTER 18

Air Conditioning

18.1. Definition and History. *Air Conditioning* is the simultaneous control of the temperature, humidity, motion, and purity of the atmosphere in a confined space. Credit for first using the expression is given to S. W. Cramer, who presented a paper on humidity control in textile mills before the National Cotton Manufacturers' Association in 1907. The term "yarn conditioning" was in common usage, and "air conditioning" seemed to be a logical expression for the process of controlling the atmospheric conditions.

When most people hear the expression "air conditioning," they think of a cool temperature in a theater or restaurant in the summer and generally regard it as a development of the 1930's. However, by definition, air conditioning applies in the heating season as well as in the cooling season, and the air-conditioning industry really obtained its start in industrial applications in the early part of this century. Since the refrigerating engineer is more interested in summer air conditioning than in winter air conditioning, the discussion in this text will be concerned mainly with the former.

Summer air conditioning includes cooling, which may be accomplished by one of several means. The best method for a given application depends upon the local atmospheric conditions; the quantity, temperature, and cost of water available; the type and cost of steam, gas, or electric power available; and the nature of the application itself. The following are the most common types of air-cooling systems:

1. Direct refrigeration system, in which the refrigerant evaporator is an air-cooling unit placed so that the air to be cooled passes directly over it.
2. Indirect refrigeration system, in which the refrigerant evaporator cools a circulating medium, such as water or brine, which in turn is used in the air-cooling unit.
3. Well-water system circulating the water through the air-cooling unit.
4. Combination refrigeration and well-water system.
5. Evaporative cooling system, in which the air-cooling unit is an air washer or other device presenting a large amount of wetted surface in the air stream and the water used is recirculated. Only enough fresh water is added to make up for that which evaporates and to flush out the system periodically.

In addition to the type of cooling equipment to use, the method of air distribution employed must be given careful consideration. Often the latter is the most critical part of the installation. Industrial applications may require an even temperature and air movement throughout. Disagreeable drafts must be avoided in comfort applications. Satisfactory methods of supplying air from either the ceiling or side wall are known. If commercial outlets are to be used, reliable data on them are available from several manufacturers. In considering the duct-sizing methods used, experience has indicated that less field adjustment and trouble are experienced when the static regain method¹ is used.

18.2. Direct and Indirect Refrigeration. Direct refrigeration is used in self-contained air-conditioning units and in most air-conditioning systems under 100 tons capacity in which the air-cooling equipment can be located near the refrigeration machine. Of course, most centrifugal compressor installations are indirect systems in which water or brine is chilled and then piped to the air-cooling units, which may be either coils or washers. With washers, little more than a 7 F rise in water temperature can be obtained, and more space and more maintenance are required than with coils. For these reasons finned air-cooling coils are now used in most new installations. If winter humidification is needed, a few small nozzles can be used effectively to spray water over the coil surfaces. A study of the economics involved and of the equipment space limitations must be made to determine whether indirect or direct systems should be used. Figure 18.1 shows a compact factory-assembled air-conditioning unit with evaporative condenser. These units are made in capacities from 3 to 40 tons.

18.3. Well Water. Well water may be used as the air-cooling medium if its temperature is low enough and if it is available in sufficient quantity. The water temperature is very important, particularly in localities where dehumidification as well as cooling is required. The water temperature must be a few degrees below the required apparatus dew-point temperature. For example, a community having 52 F well water available uses it for air-conditioning offices and sparsely populated stores but must use refrigeration in restaurants and other places with high latent heat loads.

When the water available is about 60 F, it may be economical to use the water first in an outside-air precooling coil and then in the refrigerant condenser. In some instances the water from the condenser, whether it be well water or other, is run through a coil to reheat the dehumidified air, is piped to the roof and sprayed over the desired area to reduce the heat gain, or is piped to sprinkle the lawn. Provision may be made for one or more of these additional water uses, which may be installed in

¹ Carrier, W. H., Cherne, R. E., and Grant, W. A., *Modern Air Conditioning, Heating and Ventilating*. New York: Pitman Publishing Corporation, 1940, pp. 240-243.

series, but it is recommended that by-pass pipe connections be included to provide a means of control.

18.4. Evaporative Cooling. This inexpensive means of cooling may be used in climates where the outside wet-bulb temperature does not exceed the required dew-point temperature of the supply air, as in certain parts of southwestern United States. Recirculated water is sprayed into the air, which is cooled along the wet bulb line. The heat removed in

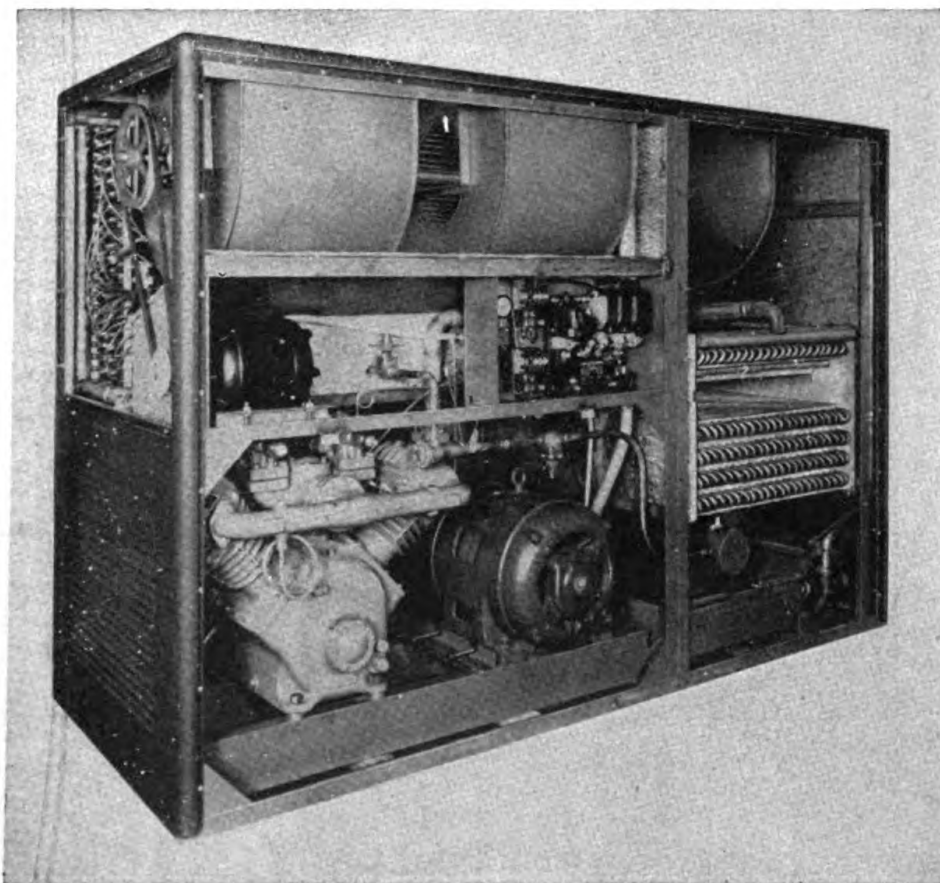


Fig. 18.1. Factory-assembled combined air conditioner, compressor, and evaporative condenser unit. Courtesy United States Air Conditioning Corp.

lowering the dry-bulb temperature of the air is absorbed by the moisture, which evaporates and raises the humidity of the air. The recirculated water remains at the air wet-bulb temperature with no external heating or cooling, and make-up water is added to replace that which is evaporated. In some units capillary tubes or shredded wood, chemically treated to prevent rot and odor, or metal wool, is used instead of spray nozzles to present the necessary water surface for air contact.

18.5. Industrial Applications. Few people realize that air conditioning is used extensively—until about 1925, almost exclusively—in manufacturing plants. Controlled air conditions are essential in the manufacture of rayon and various plastics, in color printing, in making

TABLE 18.1
REGAIN OF HYGROSCOPIC MATERIALS^a
 Moisture Content Expressed in Per Cent of Dry Weight of the Substance at Various Relative Humidities—Temperature, 75 F

Classification	Material	Description	Relative Humidity—Per Cent								Authority	
			10	20	30	40	50	60	70	80		90
Natural Textile Fibers	Cotton	American—cloth	2.6	3.7	4.4	5.2	5.9	6.8	8.1	10.0	14.3	Schloesing
	Cotton	Absorbent	4.8	9.0	12.5	15.7	18.5	20.8	22.8	24.3	25.8	Fuwa
	Wool	Austral—skein	4.7	7.0	8.9	10.8	12.8	14.9	17.2	19.9	23.4	Hartshorne
	Silk	Raw—skein	3.2	5.5	6.9	8.0	8.9	10.2	11.9	14.3	18.8	Schloesing
	Linen	Table cloth	1.9	2.9	3.6	4.3	5.1	6.1	7.0	8.4	10.2	Atkinson
	Jute	Avg—several	3.1	5.2	6.9	8.5	10.2	12.2	14.4	17.1	20.2	Storch
	Hemp	Manila & sisal	2.7	4.7	6.0	7.2	8.5	9.9	11.6	13.6	15.7	Fuwa
Rayons	Viscose	Average skein	4.0	5.7	6.8	7.9	9.2	10.8	12.4	14.2	16.0	Robertson
	Acetate	Fiber	0.8	1.1	1.4	1.9	2.4	3.0	3.6	4.3	5.3	Robertson
Paper	Newsprint	Pulp—24% ash	2.1	3.2	4.0	4.7	5.3	6.1	7.2	8.7	10.6	U.S.B. of S.
	White Bond	Rag—1% ash	2.4	3.7	4.7	5.5	6.5	7.5	8.8	10.8	13.2	U.S.B. of S.
	Com. Ledger	75% Rag—1% ash	3.2	4.2	5.0	5.6	6.2	6.9	8.1	10.3	13.9	U.S.B. of S.
	Kraft Wrapping	Coniferous	3.2	4.6	5.7	6.6	7.6	8.9	10.5	12.6	14.9	U.S.B. of S.
Misc. Organic Materials	Leather	Sole—oak tanned	5.0	8.5	11.2	13.6	16.0	18.3	20.6	24.0	29.2	Phelps
	Catgut	Racquet strings	4.6	7.2	8.6	10.2	12.0	14.3	17.3	19.8	21.7	Fuwa
	Glue	Hide	3.4	4.8	5.8	6.6	7.6	9.0	10.7	11.8	12.5	Fuwa
	Wood	Timber—average	3.0	4.4	5.9	7.6	9.3	11.3	14.0	17.5	22.0	Forest P. Lab.
	Soap	White	1.9	3.8	5.7	7.6	10.0	12.9	16.1	19.8	23.8	Fuwa
	Tobacco	Cigarette	5.4	8.6	11.0	13.3	16.0	19.5	25.0	33.5	50.0	Ford

TABLE 18.1 (Continued)
REGAIN OF HYGROSCOPIC MATERIALS²
Moisture Content Expressed in Per Cent of Dry Weight of the Substance at Various Relative Humidities—Temperature, 75 F

Classification	Material	Description	Relative Humidity—Per Cent									Authority
			10	20	30	40	50	60	70	80	90	
Food-Stuffs	White bread		0.5	1.7	3.1	4.5	6.2	8.5	11.1	14.5	19.0	Atkinson
	Crackers		2.1	2.8	3.3	3.9	5.0	6.5	8.3	10.9	14.9	Atkinson
	Macaroni		5.1	7.4	8.8	10.2	11.7	13.7	16.2	19.0	22.1	Atkinson
	Flour		2.6	4.1	5.3	6.5	8.0	9.0	12.4	15.4	19.1	Bailey
	Starch		2.2	3.8	5.2	6.4	7.4	8.3	9.2	10.6	12.7	Atkinson
Misc. Inorganic Materials	Gelatin		0.7	1.6	2.8	3.8	4.9	6.1	7.6	9.3	11.4	Atkinson
	Silica Gel		5.7	9.8	12.7	15.2	17.2	18.8	20.2	21.5	22.6	Fuwa
	Domestic Coke		0.20	0.40	0.61	0.81	1.03	1.24	1.46	1.67	1.89	Selvig
	Activated Charcoal											
	Steam Activated Sulfuric Acid H ₂ SO ₄		7.1 33.0	14.3 41.0	22.8 47.5	26.2 52.5	28.3 57.0	29.2 61.5	30.0 67.0	31.1 73.5	32.7 82.5	Fuwa Mason

² From *Heating Ventilating Air Conditioning Guide 1948*, Chapter 45.

many pharmaceuticals, in the tobacco industry, in the manufacture, development, and reproduction of photographic materials, in precision machine works for year-round production, and in the control of the quality of chocolates, certain candies, gum, matches, felt goods, several textiles, patent leather, electrical equipment, and any hygroscopic products.

Hygroscopic materials are those in which the moisture content increases or decreases when they are exposed to varying humidity conditions. Some hygroscopic materials expand and contract appreciably with changes in moisture content. One example of this is wood, as evidenced in the swelling of doors and window sashes in the humid summer atmosphere as compared with their shrinkage in the dry winter atmosphere.

Hygroscopic moisture is different from the moisture content obtained by contact with liquid water. If a wet cloth is placed in a room, it will lose moisture until it is apparently dry. The moisture retained in the cloth, which appears dry and has reached an equilibrium condition, is the hygroscopic moisture. If the cloth is heated in a drying chamber until no more moisture can be driven out, the final constant weight is called the *bone-dry weight* for the drying-chamber conditions. The ratio of hygroscopic moisture in a material to its bone-dry weight when expressed as a percentage is called *regain*. *Moisture content* is generally indicated as a percentage of the total weight and includes both hygroscopic and surface moisture. Table 18.1 shows that moisture-regain values for various materials vary widely with changes in humidity. For this reason specific standards of regain have been established for products that are sold by the pound.

Benefits to the workers' health and comfort are incidental in industrial air conditioning, but often the design conditions can be adjusted to be as near the comfort zone conditions as possible. Specific details of several particular applications will be discussed in the following sections.

18.6. Textiles. Both air conditioning and refrigeration are essential in rayon manufacture, in order to control the weight and degree of chemical reaction, to protect the worker from toxic by-products, to control the regain in the final product, to protect the product from dirt and perspiration, and to improve worker health and efficiency.³

There are not a great many rayon plants, but most of them are large and employ several large centrifugal refrigeration compressors for cooling purposes. The two chief processes used are referred to as *viscose* and *acetate* or as *wet* and *dry* spinning, respectively, both of which are 24-hr-per-day operations and require reliable air-conditioning equipment.

Recommended conditions for several textile processes are given in

³ *Refrigerating Data Book, Refrigeration Applications Volume*, 2d ed. New York: American Society of Refrigerating Engineers, 1946. "Rayon Manufacture," Chapter 74, p. 650.

Table 18.2. With the development of air conditioning it was possible to obtain these conditions and provide year-round plant operation in processing mills located near the source of the raw material. In general, an increase in relative humidity increases the length, weight, denier, elongation, softness, flexibility, pliability, and limpidity of yarns but decreases their strength. A decrease in relative humidity has the opposite effect.

18.7. Printing. Air conditioning has helped to solve some of the troublesome problems of the printer. Variations in weather cause distorting, curling, and buckling of papers, static electricity, misregister of

TABLE 18.2
TEMPERATURES AND HUMIDITIES APPLICABLE TO TEXTILE AIR CONDITIONING⁴

Process		Temperature Fahrenheit, Degrees	Relative Humidity, Per Cent
Cotton	Carding	75 to 80	50 to 55
	Combing	75 to 80	60 to 65
	Roving	75 to 80	50 to 60
	Spinning	60 to 80	50 to 70
	Weaving	68 to 75	85
Rayon	Spinning	70	85
	Throwing	70	60
	Weaving	75 to 88	60 to 75
Silk	Dressing	75 to 80	60 to 65
	Spinning	75 to 80	65 to 70
	Throwing	75 to 80	65 to 70
	Weaving	75 to 80	60 to 70
Wool	Carding	75 to 80	65 to 70
	Spinning	75 to 80	55 to 60
	Weaving	75 to 80	50 to 55
Testing	Laboratory	70	65

color printing, ink offset, ink misting, troubles with composition rolls, and distortion of wooden cut mounts. Most of these result from the reactions of the hygroscopic materials involved. Separate consideration must be given to multicolor offset lithography, sheet-fed gravure, wet-process color printing and engraving, letterpress, other sheet-fed processes, and newspaper and other web-fed printing. Control of relative humidity is most important and often results in fairly constant temperature conditions.

In multicolor offset printing it is necessary to precondition the paper properly as well as to control the atmospheric conditions. The paper must be able to lose moisture to the atmosphere at the same rate that it picks up water from the moisture used on the printing plate. Generally, preconditioning to equilibrium in a room with 5 to 8 per cent higher rela-

⁴ From *Heating Ventilating Air Conditioning Guide 1948*, Chapter 45.

tive humidity than in the pressroom is needed. A humidity of between 40 and 55 per cent may be selected for the pressrooms; 48 per cent is considered best. Control within 2 per cent relative humidity is required with a temperature variation of ± 5 F.

Sheet-fed processes require a selected humidity between 40 and 55 per cent, with 50 per cent recommended. Control within 5 per cent relative humidity and 7 F dry bulb is necessary. Web-fed processes require 55 per cent relative humidity to prevent edge cracks and static electricity. Control within 5 per cent relative humidity and 10 F dry bulb is recommended.

Storage conditions for paper are important. The installation of a good air-conditioning system in the pressroom will not necessarily solve all the problems. Paper received during cool weather should be warmed to storage-room temperature before the package is opened. After it is opened, it should be conditioned immediately for presswork if it is not already in the proper state.⁶

18.8. Candy and Gum. Before confectionery manufacturers used air conditioning, they frequently produced products of inferior quality or lost large batches when the weather suddenly changed. Often they were required to stop operations during certain seasons of the year. A sizable portion of the cooling load in a candy plant, as in a printing plant and in many other industrial plants, is the heat from power driven and gas consuming equipment.

The various types of products and processes require different conditions and methods of treatment. Usually the production manager or foreman knows what the desired optimum conditions are, and so the air conditioning engineer should work closely with him.

A constant relative humidity of 50 per cent is recommended for the starch room, where the bonbon centers for chocolates are made. If pure cane sugar is used, the temperature should be 85 F; but if glucose is used, the temperature should be 75 F. Glucose is more hygroscopic and more susceptible to higher temperatures.

Hard candy is sensitive to atmospheric conditions. The best product is obtained at about 70 F and 35 to 45 per cent relative humidity. The higher the percentage of cane sugar used, the greater the relative humidity recommended.

In making chocolate bars a rapid air circulation is needed to give a quick set and a high gloss. Cooler space conditions of 35 to 45 F and 80 per cent humidity are used. Too rapid cooling causes cracking. The receiving room from the cooler should be at 65 F and 45 per cent humidity.

Chocolate-coated candies require conditions depending upon the product composition, and slow cooling is recommended. The air dew-

⁶ *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed. New York: American Society of Refrigerating Engineers, 1946. "Printing Plants," Chapter 72, p. 638.

point temperature is important in the chocolate-dipping room for bonbons. Conditions of 61 F and 50 per cent or 67 F and 40 per cent are satisfactory.

In making gum, moisture should be evaporated from the starch at the same rate that the starch absorbs moisture from the gum; the importance of humidity and temperature control in this operation is apparent.

Candy storage rooms should be kept at 70 F and 40 per cent relative humidity. Nuts should be stored at 30 to 32 F and 75 to 80 per cent relative humidity.⁶

18.9. Low-Humidity Applications.

Drugs and Chemicals. Bottles for packaging hygroscopic products must be dried out. Relative humidities down to 5 per cent are required in some cases. Humidities below 15 per cent are more economically obtained with absorption or adsorption dehumidification equipment; otherwise, refrigeration equipment is often used.

A 70 F and 30 to 50 per cent humidity condition is required to prevent colloids from absorbing moisture and becoming sticky. Cough-drop covering should be done at 80 F and 40 per cent. Effervescent tablets need a 90 F and 15 per cent condition for manufacture. Gelatin capsules require air having not more than 0.25 grain of water per cubic foot. Glandular products and extracts must be kept below 83 F. Hypodermics are handled in atmospheres of 30 per cent humidity. Coated pills should be made in controlled atmospheres to control the product quality.

There are too many chemicals to allow consideration here of the conditions required for the manufacture of each. Their properties must be studied and experimental conditions tried for new products. Many plastics are made from hygroscopic materials that must be kept and mixed in low-humidity rooms. Figure 18.2 shows the air-conditioning ductwork in a small chemical laboratory.

Libraries and Museums. Adverse temperature and humidity conditions, as well as light, dust, acid vapors, and impurities in the materials are the main deteriorating agents to art and written records. Recommended conditions are 70 to 80 F and 40 to 50 per cent relative humidity. In addition to installations in libraries and museums, systems have been bought by individuals for their residences in order to preserve their rugs, tapestries, paintings, antiques, suits of armor, and various relics. Often the sensible heat load is small, so that reheat is needed with refrigeration systems. Chemical dehumidification systems, some with refrigeration aftercoolers, have been used for large installations.⁷

⁶ *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed. New York: American Society of Refrigerating Engineers, 1946, "Candy Plants," Chapter 73, p. 644.

⁷ *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed. New York: American Society of Refrigerating Engineers, 1946, "Libraries and Museums," Chapter 75, p. 661.

18.10. Metalworking. Machined parts requiring close tolerances, parts made in various localities or at various times of the year and which are to be assembled later, parts requiring several days to machine, and replacement parts need to be made in controlled constant temperature surroundings so as not to expand or contract unevenly. Standard test



Fig. 18.2. Air conditioning a chemical laboratory. Courtesy Consolidated Gas Electric Light and Power Co., Baltimore, Md.

gages must be kept at conditions held constant within ± 1 F and ± 5 per cent humidity, since the variation due to a 1 deg temperature change can be detected in gages. During World War II the ordnance test-gage rooms throughout the country were held at 68 F the year round, with the humidity under 50 per cent. Humidities under 50 per cent should be maintained to keep perspiration at a minimum, to prevent corrosion from handling, and also to prevent condensation on surfaces.

Deep gold, copper, and coal mines have been air-conditioned in order to have tolerable working conditions. Crane cabs that must move near or over furnaces in the steel mills have been conditioned so that an operator can work a full 8-hr shift instead of the very short shifts required previously.

18.11. Laboratories and Cabinets. Many small laboratory rooms and cabinets require closely controlled air conditions for testing or production work. Often the exact conditions desired may vary over a wide range of temperatures and humidities, requiring special equipment. Control may be obtained by overcooling and overdehumidifying, then

reheating and rehumidifying, or by controlling a three-way mixing valve in the water or brine line to the air-cooling coil.

18.12. Comfort Air Conditioning. There were only a few comfort cooling installations before 1920. Air conditioning of the Rivoli in New York City and of other theaters throughout the country in 1925 showed that summer slumps in the theater business could be overcome, indicating that similar seasonal drops in other commercial businesses could be countered. The realization by various establishments of the economic advantages of comfort air conditioning gave impetus to the growth of the industry. In many of the first installations the temperatures in the air-conditioned spaces were too low for comfort and good health. Inside design temperatures and thermostat settings were gradually raised from 70 F to within a few degrees above or below 80 F.

Personal experiences and results of investigations indicate that worker efficiency and morale as well as productivity and profits improve with the use of proper air conditioning. Merchants formerly installed air conditioning in order to draw more customers; now they must install it to keep from losing customers. The reduction in spoilage of merchandise and cleaning bills alone has made it worth while in many instances. Air conditioning has arrived, and, like refrigeration, it is here to stay. In 1946, in Baltimore alone, air-conditioning contracts (exclusive of room coolers) for over 7559 hp were let, including 6711 hp for 299 comfort installations. In Chicago in 1946 there were 517 air-conditioning installations totaling 9335 hp. A lack of equipment to meet the demand prevented the total figures from being even greater.

18.13. Stores. *Department Stores* offer a variety of problems with restaurants and lunch counters, high heat concentrations in lamp departments, variably populated basements, sparsely populated rug and furniture departments, and sun load in certain spaces. The large stores employ water-chilling refrigeration equipment; the chilled water is pumped to dehumidifying units that serve several areas, with separate fan and duct systems for each. This arrangement permits using a diversity factor so that the dehumidifying and refrigeration equipment can be smaller than the sum of the peak loads for all spaces. The equipment should be designed to produce 78 F and 50 per cent relative humidity on an average peak day. Direct-expansion refrigeration equipment with each compressor serving units on one, two, or three floors may be more economical for smaller stores. Installation costs in 1948 varied from \$1.00 to \$2.25 per square foot of floor area, with loads varying from 500 to 200 sq ft of floor per ton.⁸

⁸ *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed. New York: American Society of Refrigerating Engineers, 1946, "Department Stores," Chapter 56, p. 517.

Five- and ten-cent stores as a rule are of such size that a direct-expansion refrigeration system with one or more dehumidifying units, each serving one or more zone fan systems, works out best. People and lights cause most of the room load, with the cooking equipment contributing appreciably in the lunch-counter area. Loads vary from 350 to 120 sq ft per

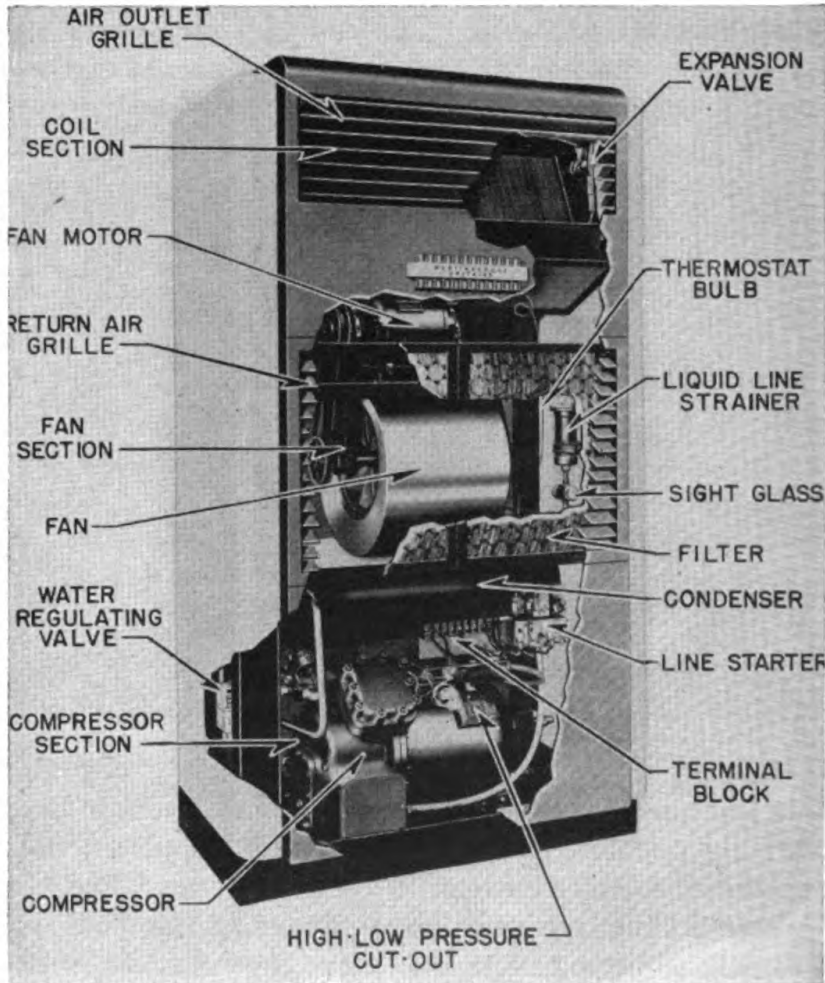


Fig. 18.3. Self-contained air-conditioning unit. Courtesy Sturtevant Division, Westinghouse Electric Corp.

ton, and installation costs in 1948 varied from \$1.25 to \$2.25 per square foot of floor.⁹

Specialty stores such as dress, jewelry, and shoe shops and drugstores of medium size are now conditioned with the large-size self-contained units having fans designed for ductwork systems. In small stores one or more self-contained "store coolers" are used more often without ductwork but sometimes with short duct systems. A typical unit is shown

⁹ *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed. New York: American Society of Refrigerating Engineers, 1946, "Five and Ten Cent Stores," Chapter 57, p. 525.

in Fig. 18.3. Outside-air duct connections should be used to provide ventilation, reduce odor concentration, and prevent infiltration. Loads vary from 350 to 110 sq ft of floor area per ton, and installation costs in 1948 varied from \$1.50 to \$3.00 per square foot.

18.14. Restaurants and Night Clubs. The cooling loads in these and similar establishments are quite variable, with the peaks often occurring at more than one time in a 24-hr period. There may be considerable heat gain from cooking equipment, and this must be separated into sensible and latent heat. As much of the cooking equipment as possible should be removed from the conditioned space to the kitchen, and the total heat output from the equipment remaining in the space can be reduced 50 per cent by adequate hooding. Separate outside-air kitchen-supply systems with filters and winter tempering coils are recommended to supply air to replace the greater part of the air exhausted from the kitchen. Failure to provide such supply-air systems has given rise to many troubles and complaints.

The high concentration of people generally produces a low sensible-heat factor, necessitating a low apparatus dew point and also requiring a high percentage of outside air to relieve the odor and smoke conditions. An exhaust system to draw air off the ceiling is recommended where there is considerable smoking. Awnings, or at least Venetian blinds, should be used on sunlit windows. Door infiltration should be considered, and a vestibule or revolving doors should be installed if they would reduce the load appreciably.¹⁰

Activated carbon may be used to advantage to absorb odors and reduce the outside-air requirements if smoking is not excessive or if the smoke can be effectively removed by exhausting less air than is brought in from the outside. Dehumidification by absorption or adsorption equipment has been used in some instances. Self-contained air-conditioning units with or without ductwork are used in small establishments.

Design loads vary from 135 to 75 sq ft per ton, and installation costs in 1948 were between \$2 and \$5 per square foot of floor area.

18.15. Multiroom Buildings. These include office buildings, hotel guest rooms, apartments, hospital private rooms and wards, and passenger-ship staterooms. Most of the rooms have one or more outside exposed walls so that the sun load must often receive careful consideration and study. The light load may also be of considerable magnitude, and the question then arises whether the lights will be on or off when the sun load occurs. The lights are seldom turned off in offices where artificial light is required for part of the day, but they are usually turned off in the rooms of other buildings when sun shines through the windows.

¹⁰ *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed. New York: American Society of Refrigerating Engineers, 1946, "Eating and Amusement Places," Chapter 59, p. 544.

Since the solar heat gain is constantly shifting, ships and buildings having rooms with different exposures must be zoned. This means that at least one separate duct system with its own control should serve those rooms exposed to the east and likewise for those to the south, to the west, to the north, and also for the interior rooms. Provision should be made to meet the critical sunny fall and spring days when heating is required for the north rooms and cooling for the south rooms.

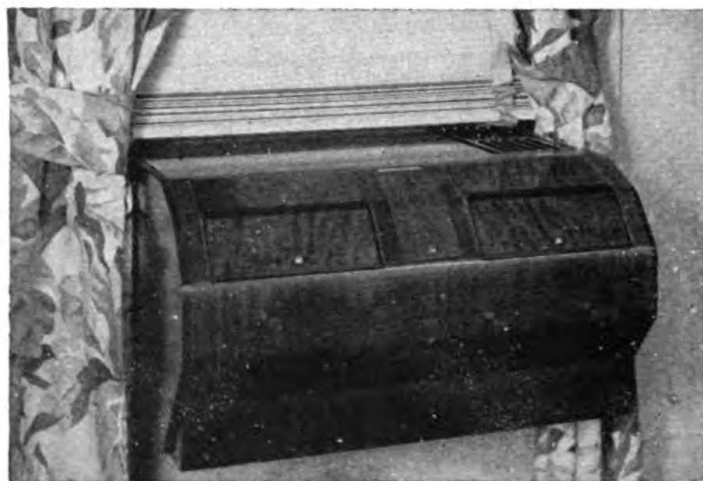


Fig. 18.4. Window-type room cooler. Courtesy York Corp., York, Penna.

Heat from hot-water pipes, from internal equipment, and from adjoining warm spaces must not be overlooked. In accordance with the latest practice, the outside-air quantity is computed from the values in Table

TABLE 18.3
VENTILATION STANDARDS¹¹

Type of Building	Outside cfm per sq ft
	Floor Area
Office buildings.....	20-25
Apartments.....	25-30
Hotel guest rooms.....	30-35
Hospital patient rooms.....	30-35

18.3. These values are satisfactory, since there are few or no people smoking in the densely populated areas and the greatest amount of smoking is done in the large private offices.

In new buildings individual units are being installed in the small and private offices and supplied with dehumidified outside air at conventional velocities from ducts below or in the floor or from vertical high-pressure, high-velocity conduits of 4 to 8 in. diameter. In either case recirculated

¹¹ From *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed. New York: American Society of Refrigerating Engineers, 1946, "Multi-room Buildings," Chapter 60, p. 554.

air from the room is induced over a water coil and mixed with the high-pressure jet of the primary air and discharged back into the room. If only a few small offices are to be cooled, self-contained room coolers of $\frac{1}{2}$ to $1\frac{1}{2}$ tons such as shown in Figs. 18.4 and 18.5 are used.

Loads for office buildings vary from 500 to 200 sq ft per ton, and 1948 installation costs ranged from \$1.25 to \$4.50 per square foot of floor area. For apartments and hotel rooms the values are 900 to 400 sq ft per ton.

18.16. Amusement Spaces.

Theaters. Many changes in equipment and system design for theater air conditioning have been made since the Los Angeles Metropolitan



Fig. 18.5. Console-type room cooler. Courtesy York Corp., York, Penna.

Theater installation in 1923. Enough experience has been gained to formulate standard recommendations.

The cooling load consists mainly of the occupants and of the design outside-air quantity, which is based upon the occupancy. The average man gives off 400 Btu per hour but the average woman only 350 Btu per hour and the average child 270. Thus, if the audience consists of an appreciable percentage of women and children, the average heat gain per occupant may be as low as 350 Btu per hour. The minimum outside-air quantity used when smoking is not allowed is 5 cfm per person, but this usually will not provide enough dilution to reduce the odor concentration below that generally preferred. A value of 6.25 cfm per occupant, including standees, is recommended, but the value actually used depends upon the type and location of the theater.

Inside design conditions of 78 to 80 F with 55 per cent relative humidity are recommended. The apparatus dew point will then be slightly

above 53 F. Downward diffusion or horizontal diffusion of the supply air may be used. In horizontal diffusion, blowing from the side or from rear to front is preferred, and 15 to 18 cfm per occupant is sufficient. About 25 cfm per occupant is required with downward diffusion to prevent cold spots. As in other applications, provision should be made for handling all the air from the outside in mild weather in order to save on operating costs. Incidentally, the installation of radiation at all doors and on the stage is recommended.¹²

Total loads vary from 20 to 15 seats per ton, and installation costs in 1948 ranged from \$15 to \$30 per seat.

Bowling Alleys. Air conditioning has also made it possible for these establishments to function the year round. When estimating the load, 490 Btu sensible heat and 1010 Btu latent heat per bowler is used, but only one person per alley can actually bowl at any one time. Only the occupied space is conditioned, and air is distributed either from the side or toward the bowler from a short distance down the alleys.

Dance Halls and Skating Rinks. An adult actively dancing or skating produces about 450 Btu per hour sensible heat and about 940 Btu per hour latent heat. It should be remembered, however, that all those present are not actively participating all the time.

18.17. Transportation.

Trains. The railroad industry was the first in this classification to use air-conditioning equipment, initially installing it in about 1930. Three methods of refrigeration are used at present: ice bunkers, mechanical compression, and steam ejection. Initial cost of the ice system is low, but operation cost is high, so it would be most practical where the season is short and ice is inexpensive, as in Canada. Costs of the other two systems are comparable. Electric drive, internal-combustion-engine drive or direct-shaft drive may be used for compression systems. Each system has certain advantages and disadvantages.¹³ The cooling load per car is from 6 to 8 tons, and 2000 to 2400 cfm with 25 per cent outside air is circulated. Typical refrigeration and related equipment installed in a train built in 1947 is shown in Fig. 18.6.

Airplanes are continually being changed in design and operation, so that any specific information given may soon be out of date. It appears that the practical way to absorb the heat of pressurization is by air-to-air refrigeration. Light, high-speed 5-ton units seem to be preferred. Since the load while on the ground greatly exceeds that in flight, mobile

¹² From *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed. New York: American Society of Refrigerating Engineers, 1946, "Theaters," Chapter 61, p. 573.

¹³ *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed. New York: American Society of Refrigerating Engineers, 1946, "Passenger Cars," Chapter 63, p. 596.

self-contained units or underground units at loading locations are connected to commercial planes that are awaiting takeoff.^{14,15,16} (See §7.10.)

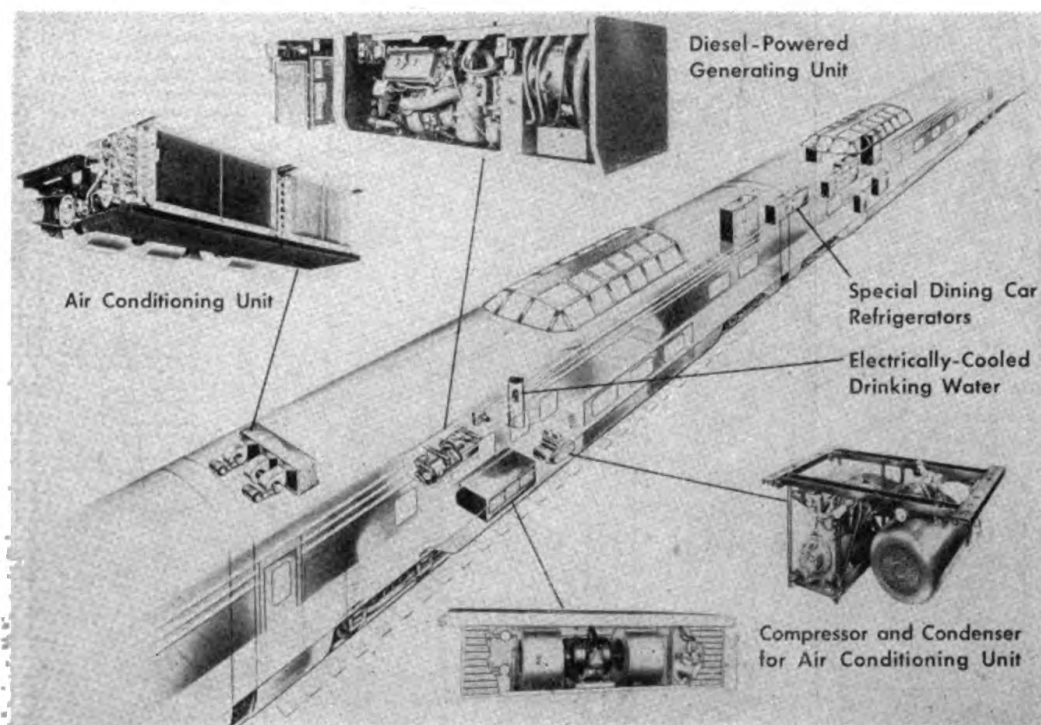


Fig. 18.6. Passenger-train refrigeration. Courtesy Frigidaire Division, General Motors Corp.



Fig. 18.7. Combination supply- and return-air ceiling unit. Courtesy Consolidated Gas Electric Light and Power Co., Baltimore, Md.

Buses and trolleys are new applications. Air conditioning was first applied to fleet operation in 1938, but by 1942, 75 per cent of all intercity

¹⁴ *Ibid.*, "Airplanes," Chapter 66, p. 606.

¹⁵ "Air Conditioning a Luxury Airliner." *Heating and Ventilating*, Vol. 43, No. 10 (October, 1946), p. 95.

¹⁶ "Refrigeration for Air Conditioning Aircraft." *Heating, Piping and Air Conditioning*, Vol. 18, No. 4, p. 79.

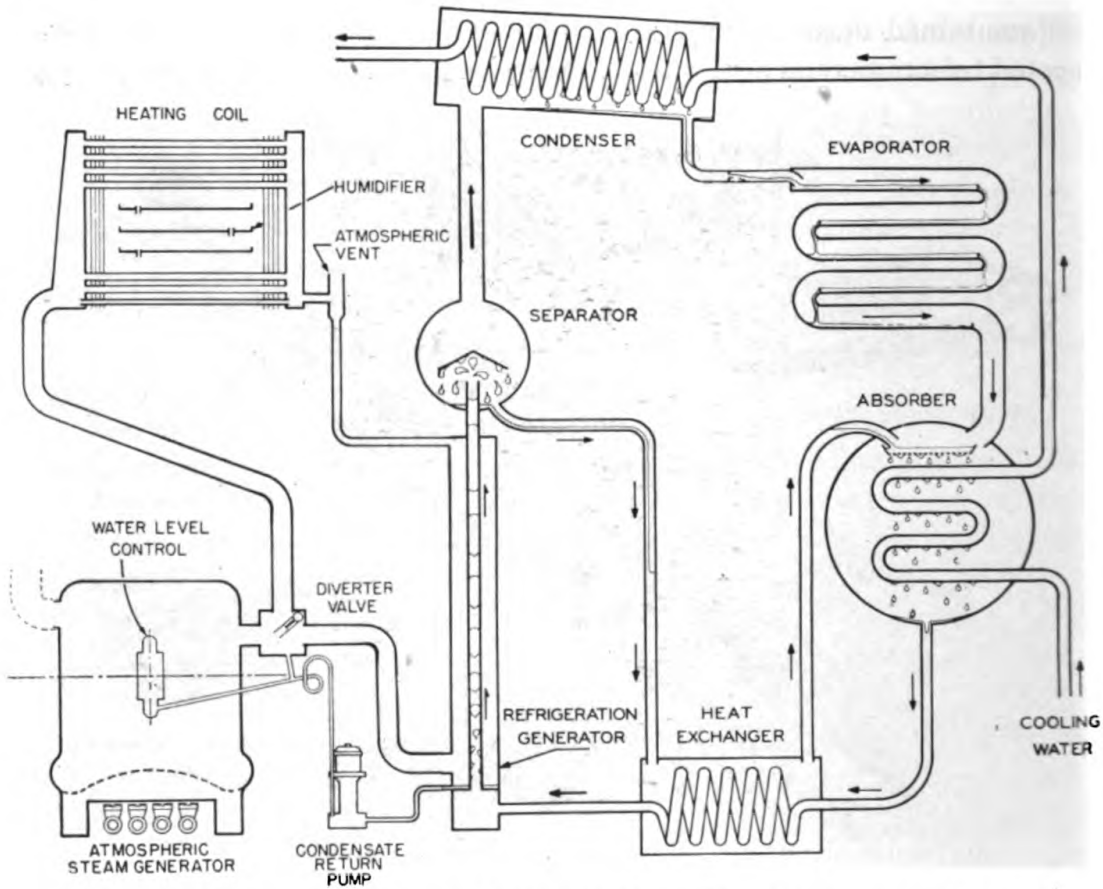


Fig. 18.8. Diagram of all-year air-conditioning cycle. Courtesy Servel, Inc.

buses under construction were air-conditioned. Mechanical compression units are being driven from the main engine or by a separate gasoline engine. Present equipment weighs from 1000 to 1200 lb, or 20 per cent of the total passenger load. Capacities range from 3.5 to 4 tons for 35- to 40-passenger buses. Heat-absorbing glass and double sash are desirable, but the latter adds weight. Air supply through a perforated ceiling is probably the best procedure.¹⁷ Greyhound started to put in operation 1500 new air-conditioned 37-passenger buses in 1947 and built an experimental 50-passenger double-decker for long runs. Design details for air-conditioning 36 and 41-passenger buses have been published.¹⁸

• An air-conditioned trolley coach in Atlanta was so successful that 100 additional coaches were ordered; some details of this venture have been published.^{19,20}

¹⁷ *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed. New York: American Society of Refrigerating Engineers, 1946, "Buses," Chapter 64, p. 601.

¹⁸ "Air Conditioned Buses," *Heating and Ventilating*, Vol. 43, No. 2 (February, 1946), p. 80.

¹⁹ "Results of Studies of Air Conditioned Trolley," *Heating and Ventilating*, Vol. 43, No. 9 (September, 1946), p. 92.

²⁰ "Cooled Trolley Coaches," *Heating and Ventilating*, Vol. 44, No. 4 (April, 1947), p. 116.

18.18. Residences. The potential market for residential air conditioning is tremendous. The Edison Electric Institute reported in 1942

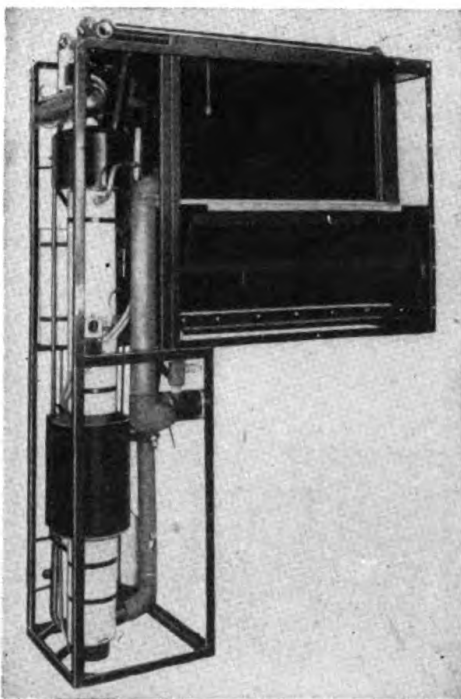


Fig. 18.9. Absorption refrigeration equipment. Courtesy Servel, Inc.

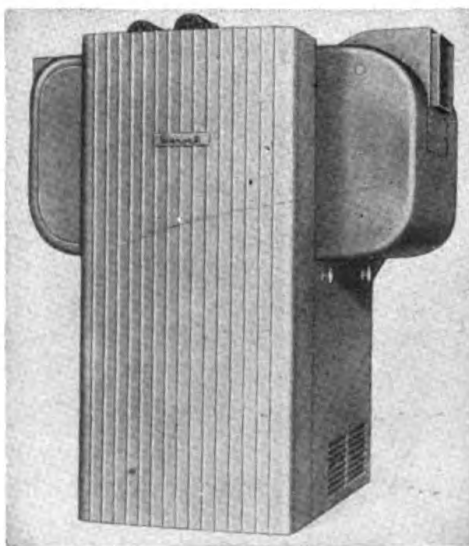


Fig. 18.10. All-year air-conditioning unit. Courtesy Servel, Inc.

that only 22,000 of 21,000,000 urban homes with electricity had air-conditioning installations, and new homes are being built as fast as possible since World War II. The cost of the equipment still remains as the unsolved problem and presents the greatest obstacle to widespread usage.

There is a challenge to engineers to meet and solve this by new application, design, or production procedure.

Most of the heat gain is from transmission and solar heat, which presents a control problem in large houses as the sun load shifts during the

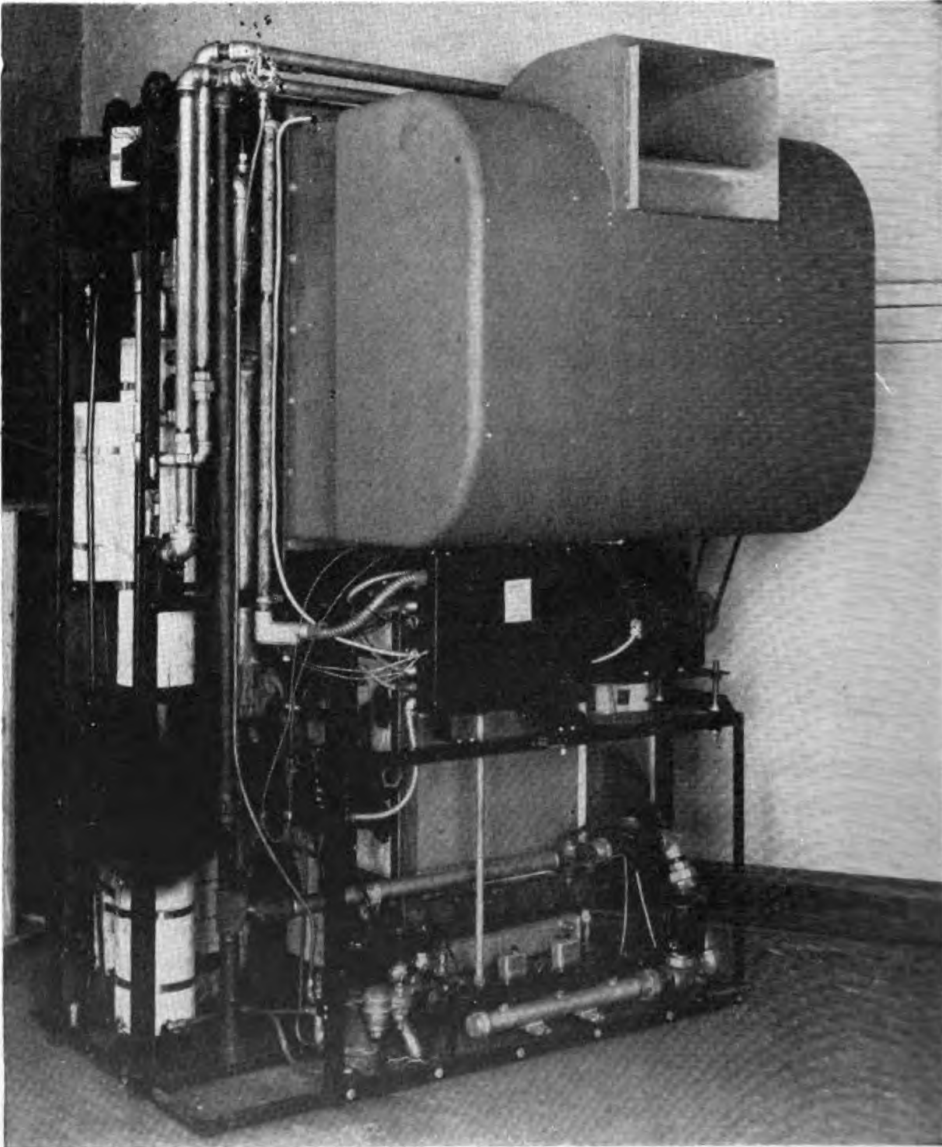


Fig. 18.11. All-year air-conditioning equipment. Courtesy Servel, Inc.

day. It is necessary to install refrigeration capacity only for the **maximum** load for the house as a whole and not for the total of the **maximum** room loads. With forced-air systems high-side-wall supply registers are preferred, and with proper location of these and of return grilles, the **same** system may be used for heating. If the ductwork can be located **above** the ceiling, combination supply and return-air units can be used for cooling, as shown in Fig. 18.7.

Self-contained room coolers such as are shown in Figs. 18.4 and 18.5 are sometimes used if it is not desired to cool the entire house. Year-round gas-fired units using absorption refrigeration have been developed in recent years, as well as the electric-driven heat-pump units for the central-type system. The schematic diagram of an all-year gas-fired air-conditioning unit is shown in Fig. 18.8 and the refrigeration equipment only in Fig. 18.9. The external casing is shown in Fig. 18.10 and the entire unit in Fig. 18.11. More air is desired for cooling than normally is used in forced-air heating, so that larger ducts should be used. Satisfactory results can be obtained in some cases by installing a cooling coil in an existing forced-air heating system.

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4. *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed. New York: The American Society of Refrigerating Engineers, 1946.

PROBLEMS

18.1. A quantity of skein wool weighs 100 lb after reaching equilibrium at 75 F and 80 per cent humidity. How much will it weigh after equilibrium if removed to a 20 per cent humidity room?

18.2. How much will kraft wrapping paper weigh at 80 per cent humidity per pound of paper at 10 per cent humidity?

18.3. Two adjoining spaces each of 3000 sq ft floor area are to be air-conditioned. One is to be a restaurant and bar, the other a ladies' dress shop. What would be the probable range of refrigeration tonnage for each and the probable range of cost of air conditioning at 1948 prices?

18.4. Calculate the total cooling load for the following office in Philadelphia:

Inside conditions: 78 F dry bulb, 66 F wet bulb	
Net east wall, 12-in. brick, $\frac{1}{2}$ -in. plaster	99 sq ft
Net east glass with inside Venetian blinds	48 sq ft
Gross window area	58 sq ft
Partitions, 4-in. tile, plastered each side	442 sq ft
Glass partition	93 sq ft
Floor, asphalt tile, 4-in. concrete furred and plastered	270 sq ft
Ceiling, same as floor	270 sq ft
Occupants	3
Adjoining spaces: nonconditioned offices	

18.5. (a) Using the median value from the range of values given for each case in the text, determine the floor area or number of seats that a 50-ton air-conditioning system would serve for each of the following: restaurant, dress shop, five-and-ten-cent store, and theater. (b) About how much would each system cost at 1947 prices?

CHAPTER 19

Food Preservation

19.1. General. Although "man shall not live by bread alone," food and drink are necessary, and refrigeration is essential in many phases of the preparation, storage, and distribution of various foods and beverages. The primary purpose of food refrigeration is to aid preservation, which has contributed greatly toward increasing the standard of living as well as reducing wastage.

There are four distinct scientific approaches to the problem of food preservation: heat treatment as applied in canning, drying or dehydration, physico-chemical modifications, and refrigeration. The primary purpose in each of these methods is to arrest the growth, to inhibit the action, or actually to destroy the microbiological and enzymic agencies that eventually would break down, deteriorate, or spoil the food substance. All are based on scientific principles which can be understood or applied most effectively after studying the results of the research work that has been conducted in that particular field.

Nicolas Appert has been credited with the discovery of canning in 1809 in France. He won great acclaim and was awarded 12,000 francs by the emperor Napoleon for his crude method of preserving food in bottles. Canning continued to be the most popular method of preserving food in small quantities for over a century, but modern refrigeration is the most promising for preserving foods in their natural state for comparatively long periods of time. "And few patrons want to go back to canning and pickling after they have eaten frozen food."¹ Food preservation was one of the first applications of mechanical refrigeration; even frozen meat was successfully shipped from the United States to England as early as 1875. The recent great expansion in this field probably will continue, since it is being constantly aided by research and development work.²

The refrigeration method of preserving is applied in various ways to different products. Chilling of a fruit or vegetable retards its rate of maturing and thus may prolong its life. Keeping a product at a reduced temperature also retards the spread of bacteria that would cause the product to spoil if allowed to become active. Many food products can be preserved and made to retain their original flavor and texture by proper

¹ "Freezer Lockers." *Fortune*, Vol. 36, No. 3 (September, 1947), p. 198.

² Prescott, S. C., "The Progress of Research in Refrigeration." *Refrigerating Engineering*, Vol. 53, No. 1 (January, 1947), p. 15.

freezing. Several of the applications of refrigeration in food preservation will be explained in the following sections.

19.2. Food Freezing.

Theory. The practical benefits of extremely fast over slow freezing rates have been clearly demonstrated. The advantages are attributed to phenomena as explained by two theories, both of which probably apply.

The *mechanical theory* points out that plant and animal tissues consist of many cells containing and surrounded by a liquid having various types of substances in solution or suspension. Among the changes that take place during cooling below 32 F is the formation of ice crystals. Most of the crystals form between 31 and 25 F, but small amounts may continue to freeze out until -70 F is reached. The amount of ice formed at any given temperature seldom varies with different chilling rates, but the size of the ice crystals varies inversely with the speed of freezing. The effect is similar to that of a quenched steel. Slow cooling produces fewer and larger crystals, which, as they develop and grow, damage the cell tissues. Upon thawing, the substance with damaged tissues loses its original texture and tasty juices. Since animal cell walls are more elastic than fruit and vegetable cell walls, the former are not broken so easily, and their freezing rate is not so critical.

The *physico-chemical theory* asserts that during the formation of ice crystals the remaining solution continually becomes more highly concentrated with the dissolved salts. This concentration causes a chemical salting-out effect so that the soluble proteins become less soluble and may precipitate out. The concentrated salts also tend to dehydrate the protein by osmosis. The protein is thus irreversibly denatured and cannot absorb its original moisture, and the original flavor may be lost. Fast freezing may congeal the colloidal sol and afford a protection to prevent the loss of water by osmosis from the colloid.³

Methods. It is difficult to classify distinctly the many processes used for freezing, since the principles applied often overlap. The principles used include convection freezing with air that has been cooled by pipe coils, plates, or unit coolers and applied in sharp freezers; air blasts in tunnels where the product is conveyed on a belt, on trays, or on trucks as shown in Fig. 19.1; direct contact with an edible liquid coolant by spraying it over the product or immersing the product in it; indirect contact by placing the product on or against a chilled metal surface; and the creation of a high-vacuum atmosphere by steam nozzles so that the absolute pressure corresponds to a saturation temperature well below freezing.⁴

³ *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed. New York: American Society of Refrigerating Engineers, 1946, "Theories and Methods of Quick Freezing," Chapter 1, p. 5.

⁴ "Steam Ejector 'Flash Freezes' Foods." *Air Conditioning and Refrigeration News*, Vol. 50, No. 2 (January 13, 1947), p. 14.

The convection principles are usually employed in the freezing room of locker plants, although refrigerated plate shelves are used in some. Most domestic units used for freezing rely on a combination of convection and indirect contact. These as well as the other principles are applied in the systems of commercial concerns that freeze large quantities of products. Sodium chloride brine is used in the spray systems, and either sodium chloride brine or sugar sirups are employed in immersion systems. Descriptions, disadvantages, and advantages of the various systems are

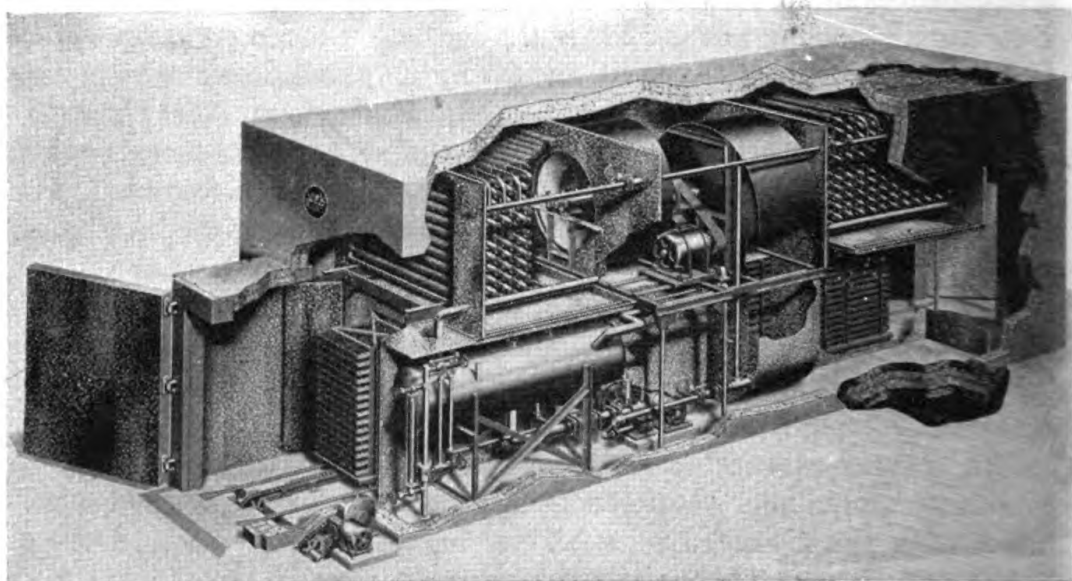


Fig. 19.1. Freezing unit for food conveyed on trays or trucks. Courtesy York Corp., York, Penna.

given in the literature.⁵ Relative performance of 21 commercial food-freezing systems have been analyzed by Meek and Greene⁶ and by Hulse.⁷

The time required for freezing a number of different products by -20°F blast freezing as reported by Pollock is shown in Table 19.1.⁸

19.3. Preparation for Freezing. Anyone contemplating the quick-freezing of food products should understand the similarities and differences between animal, fruit, and vegetable tissues. Because of these differences, various procedures before and during quick-freezing operations are required for unlike products. Animal tissues except oysters and clams are dead when they reach the freezing plant, but plant tissues remain alive until blanched or frozen. The comparatively inelastic plant cells require faster freezing rates than animal tissues, although some

⁵ See Bibliography at end of chapter.

⁶ Meek, G. W., and Greene, van R. H., "Evaluating Food Freezing Methods." *Refrigerating Engineering*, Vol. 53, No. 5 (May, 1947), p. 391.

⁷ Hulse, C. W., "Quick Freezing Plants and Their Operation." *Ice and Refrigeration*, Vol. 112, No. 5 (May, 1947), p. 43.

⁸ Pollock, E. O., "Blast Freezing Plants." *Refrigerating Engineering*, Vol. 48, No. 1 (July, 1944), pp. 16-19.

TABLE 19.1
BLAST FREEZER PERFORMANCE WITH -20 F AIR⁹

Product	Weight of Container	Depth of Product in Container, in.	Time Required to Reduce to 0 F, min
1. Peas.....	38 lb tray	1½	50
2. Peas.....	23 lb tray	¾	35
3. Corn.....	44 lb tray	1½	90
4. Corn.....	24 lb tray	¾	40
5. Carrots.....	38 lb tray	1½	50
6. Ground meat...	100 lb pan—reduced to 20 F in 16 hr		
7. Boneless meat...	100 lb box—reduced to 0 F in 25 hr		
8. Boneless meat...	50 lb box—reduced to 0 F in 16½ hr		

Product	Time Required to Reduce to 0 F, hr
9. Veal chuck, open blast.....	4½
10. Lamb carcass, open blast.....	6
11. 14 lb hams, open blast.....	6 to 8
12. 5 lb block shrimp, pan.....	3 to 4
13. 5 lb block fish fillet, pan.....	2 to 3
14. 5 lb dressed salmon, open blast.....	1½
15. 1 lb box chicken a la king.....	1½
16. 35 lb wood, closed box poultry.....	12
17. 36 lb closed carton poultry, 3½ in.....	9½
18. 22 lb open carton poultry, 3½ in.....	4
19. 22 lb closed carton poultry, 6 in.....	7½
20. 5 lb block backed beans, pan.....	2½
21. Turkey, open blast.....	5½

fruits may be mixed with sugar and frozen slowly to allow the sugar to penetrate.¹⁰

Enzymes, which are complex organic substances capable of transforming by catalytic action some other compound, start affecting fruits and vegetables directly after harvest. Their reactions must be constantly controlled or they may progressively damage the product until it is cooked or eaten. The addition of sugar to certain fruits before freezing retards any enzyme action and also lessens the leakage of juices during thawing. Vegetables are subjected to some form of heat such as blanching, a quick scald with hot water or steam, just before freezing in order to stop enzymic changes during storage.

The three causes of fish spoilage are (1) action of bacteria acquired

⁹ From Pollock, E. O., "Blast Freezing Plants." *Refrigerating Engineering*, Vol. 48, No. 1 (July, 1944), p. 19.

¹⁰ *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed. New York: American Society of Refrigerating Engineers, 1946, "Fruits & Vegetables," Chapter 2, p. 10.

by handling out of water; (2) oxidation of the oil or fat in the flesh; and (3) enzyme action in the tissue which becomes destructive after death. Storage at as low a temperature as possible before freezing, freezing as soon as possible, providing a lasting glazed ice coating over each piece, and, finally, storage at subzero temperatures will keep the three deteriorating actions at a minimum.¹¹

The important factor in meat preparation is to carry out both the complete slaughtering operation and the transfer of meat into a cooler as quickly as possible. This procedure allows a minimum time for exposure to temperatures that promote bacteria growth and color deterioration. Beef should be aged to improve its tenderness, cuts for steaks and dry roasts requiring a longer period than other cuts. Packing plants either age their beef about three weeks at 32 to 36 F and 86 to 92 per cent relative humidity or for a shorter period at 45 to 60 F with germicidal lamps. The tough tissues are broken down by enzyme action, which is faster at higher temperatures. Aging of meat to be frozen should not be over 10 or 12 days at the low temperature. Longer aging decreases the storage life; furthermore, it is thought that freezing aids tenderizing.¹²

Product dehydration must also be prevented. To accomplish this end, all products must be placed in sealed moistureproof containers directly after freezing if they were not placed in them before freezing. Only small-sized products that can freeze completely through quickly and can be frozen by spraying, immersion, or very low-temperature tunnel or other air-blast-type systems are not prepackaged. Most meats are packaged before freezing even though immersion or air-blast methods are used. Various packaging materials have been tested.¹³

Meats should first be cut to the smallest size desired for the final cooking before eating. Fruits and vegetables should also be frozen in as small a package as practical, although this practice is not so critical for well-sugared fruits. The larger the package, the longer it takes for the heat to be removed from the center, and the freezing time is prolonged.

Freezing will not improve a product. Therefore only the best quality products, including limited varieties of vegetables, should be frozen. Lists of the latter are available, together with considerable other detailed information on preparation and procedures for freezing foods.¹⁴

¹¹ *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed. New York: American Society of Refrigerating Engineers, 1946, "Fish Refrigeration," Chapter 3, p. 15.

¹² *Ibid.*, "Freezing of Meats," Chapter 5, p. 43.

¹³ McCoy, D. C., Cook, S. V., and Hayner, G. A., "Study of Frozen Food Wrapping Materials." *Refrigerating Engineering*, Vol. 52, No. 6 (December, 1946), p. 531.

¹⁴ *Refrigerating Data Book, Refrigeration Applications Volume*, 2nd ed. New York: American Society of Refrigerating Engineers, 1946, "Preparation of Fruits and Vegetables for Home Freezing," Chapter 7, p. 66.

In addition, the national and state departments of agriculture have many available publications on the subject, and several books have been published (see Bibliography at end of chapter).

In recent years considerable research and experimenting has been conducted on frozen precooked foods and completely ready-to-cook foods such as pies and other bakery goods. The housewife with a freezing unit can prepare foods in larger than normal quantities and freeze some for future use. Famous tasty dishes can be prepared, frozen, and shipped to her from New Orleans, San Francisco, Boston, or wherever they originate. Immediate freezing followed by storage below 0 F until the buyer is ready to thaw and prepare the food for eating is essential.

19.4. Frozen-Food Storage. The storage-room conditions for frozen foods are as important as the selection and preparation of the product and the actual freezing process. A temperature between 0 and -10 F is considered most desirable, although some sugared fruits are kept best at -15 F. Fluctuations in storage temperatures are more serious than the actual temperature selected and are the chief cause of most troubles. If variable temperatures are to be encountered, the selection of a lower storage temperature is recommended to lessen the harmful effects. Storage spaces in which products are to be frozen as well as stored should be loaded with only a small quantity of unfrozen food at any one time. Also, the product being frozen should be prechilled as close to 32 F as possible and then not placed near the previously frozen products.

A high relative humidity, 85 to 95 per cent, is desirable, although this condition is not vitally important if all products are in high-quality moistureproof packages or containers. It may be necessary to reglaze unwrapped fish and other ice-coated products to prevent dehydration after evaporation of the previous glazing.

19.5. Locker Plants. These establishments were first built in the Pacific Northwest and developed into very worth-while projects in the 1930's, particularly in Iowa and the surrounding states. With almost 700 plants in Iowa in 1947, there is hardly a city, town, or village of any size in that state that does not have at least one locker plant. Plants are also being installed in other parts of the country at the rate of 700 per year in 1947, when the total number of installations was about 8700. Everyone benefits from a locker plant—the equipment manufacturer, the refrigeration dealer, the building contractor, the plant owner and operator, the patrons, and the other local merchants. The locker plant may be a separate enterprise or a part of a grocery store, meat market, dairy, or other establishment. It consists of four or more distinct spaces.

The fresh-killed meat and other food products are first placed in the *chilling and aging room*, which is designed for 35 F. Its size depends upon the types of products and the rate of processing; the latter is a function of the number of lockers. The incoming product averages about 2 lb per locker per day, but the chill-room space and equipment should be designed for a minimum of 3 lb per locker per day in order to handle the variation from the average load.

A *processing room* is needed, in which the products are prepared and packaged for freezing. Meats are cut into the desired roasts, steaks, chops, or other cuts, and fowl are cleaned. They are then placed in moistureproof packages and identified. The patrons are charged a few cents per pound for having their meat chilled, cut, wrapped, marked, and frozen. The processing room is not refrigerated.

A separate *freezing cabinet or room* is provided for freezing the packaged products before they are placed in a patron's locker. As in the chill room, the freezer space and equipment are designed for freezing about 3 lb per locker per day. The design temperature should be from -10 to -20 F.

The *locker room* contains the patrons' lockers, which range in size from 5.5 to 7.5 cu ft of space. They are usually in rows of tiers separated by aisle spaces, with five or six lockers per tier. The upper two or three are door-type, and the lower two or three are drawer-type. A 6-cu-ft locker holds about 150 lb of meat and rents for \$10 to \$15 per year. The room design temperature should not be above 0 F.

The size of locker plants varies from less than 100 to over 1500 lockers. The larger plants often have other rooms in addition to the four essential ones. Sometimes the chill room is separated from the aging room, and a curing room is added. A smoke room, smoke cabinet, lard room, fruit and vegetable room, and others are added in larger plants.

In general, one compressor is used for the chill, age, and cure rooms, which are kept near 35 F, and another compressor is used for the locker and freezer rooms, which are kept at 0 F or lower. Plate evaporators or unit coolers are used with Freon-12 in most plants. Frost may be scraped off the plates, but water or other defrosting methods are needed for the unit coolers. Some patented systems have been installed in which the patron does not need to be in a 0 F room in order to have access to his locker. A suggested design for a medium-sized plant is shown in Fig. 19.2. Possibilities of using the condenser heat for the office and processing rooms in winter might be worth considering.

19.6. Cold Storage. Many food products may be stored at temperatures above freezing, although the length of storage time is considerably less than when the product is in the frozen state. Maximum length of storage as well as the optimum conditions for each product are given in Table 10.2. Higher temperatures can be tolerated for shorter storage periods. A proper humidity condition is usually as important as temperature, and some products are more susceptible to air movement than others. Each product has certain peculiarities that require special precautions in order to reduce spoilage. Research has provided the answers to many problems, and it is recommended that the available literature be studied before definite plans are made for storing any sizable quantity of a product for any length of time. Not only fresh produce but also

dehydrated eggs, meat, milk, and vegetables can be kept satisfactorily for a longer time at reduced temperatures.

In planning a cold-storage plant careful study should be given to the size and the location. Conveniences and transportation facilities must be considered, as well as the internal design and type of refrigerating plant. Compound ammonia refrigerating machines chilling brine in two separate

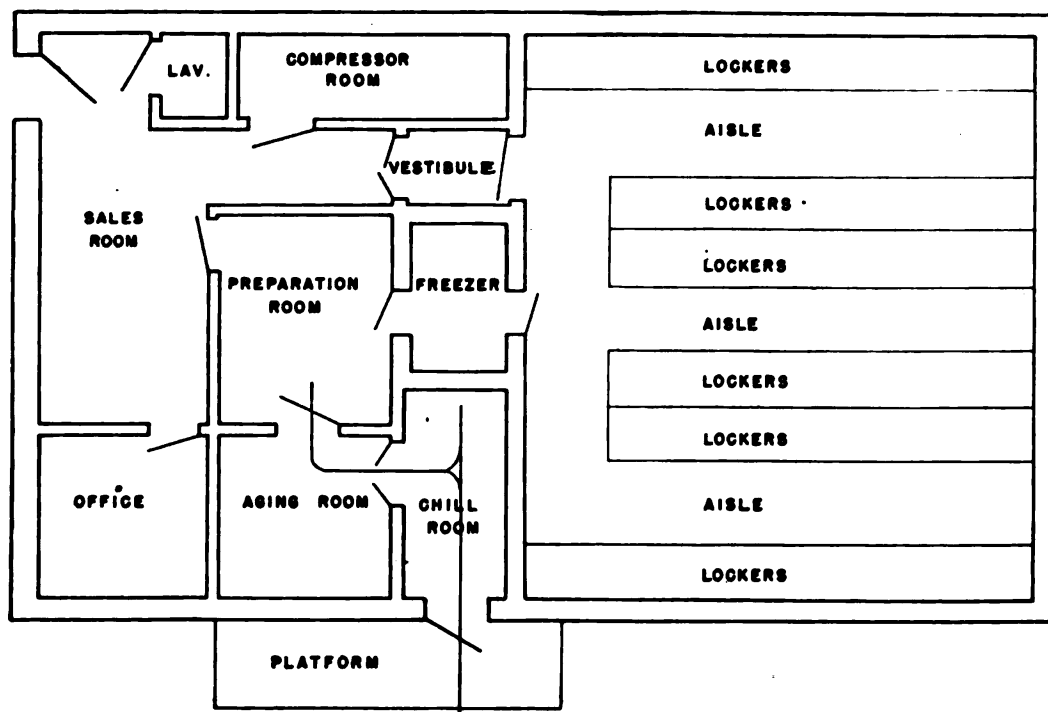


Fig. 19.2. Suggested locker-plant layout.

systems, one for subfreezing requirements and the other for rooms kept above freezing, have been used in the past.

19.7. Prepackaged Fresh Produce. Considerable interest was developed in 1946 in prepackaging fresh fruits, vegetables, and even meats in consumer units. This project created new problems in refrigeration applications, on which certain experimental work has been done.¹⁵ The produce is usually received at the warehouse in the evening, placed in the cooler, packaged the following day, and returned to the cooler, where it remains until loaded on the delivery truck the next morning. Considerable spoilage can be eliminated in this way if packaging is done quickly and properly. It also saves time for the grocer and shopper. However, better refrigeration than was first used is needed. Possibly provision for more air circulation around the packages, in order to improve the cooling of those near the center, would help.

¹⁵ Bratley, C. O., "Refrigeration of Prepackaged Fruits and Vegetables." *Refrigerating Engineering*, Vol. 52, No. 6 (December, 1946), p. 516.

19.8. Food Processing. Ice cream was one of the first processed foods produced commercially that required refrigeration. Various claims have been made as to where and when the ice-cream industry started. However, the first reference to the commercial distribution of ice cream appearing in Baltimore was in an advertisement for Mr. DeLoubert's recreation and confectionary parlor in the June 15, 1797, issue of the *Maryland Journal and Baltimore Advertiser*.¹⁶ Vallette and Company announced in the *Federal Gazette* of May 15, 1798, that they could "supply ice cream thereafter to the ladies and gentlemen of Baltimore." River or lake ice and salt were used for cooling, and ice cream remained a luxury item until more than a century later, when mechanical refrigeration made large-scale production possible.

In the modern ice-cream plant, refrigeration is used for cooling the room where the ingredients are stored, for cooling from 150 F to 40 F the mix leaving the homogenizer, for freezing the mix and chilling it to between 24 and 21 F, and for the -10 to -15 F hardening room, where the containers are placed immediately after being filled with the mix from the freezer.

Refrigeration is needed for several processes in dairies. Raw milk, cream, and sweet cream for butter should be kept at 40 to 50 F before pasteurization. Milk and cream must be chilled quickly after pasteurization. Chilling of butter, curing of cheese, and the storage of all products before delivery require refrigeration. Chilled water, brine, and direct-expansion systems are all used.

Bakeries require many forms of refrigeration systems. The ingredient storage room and fermentation room must be cooled; chilled water or ice may be needed; the dough mixer requires cooling; chocolate and other special cookie rooms need controlled air conditioning; and air conditioning the packaging room reduces mold spores, bacteria, and dust as well as increases worker efficiency.

Breweries use carefully controlled temperature conditions for each step in making beer or ale. The hop storage room and yeast storage room must be cooled, and cooling is also required during each process from the time the wort leaves the brew kettle until delivery, except that the bottled and canned products must be pasteurized as a part of the final process. Controlled enzyme action is utilized to promote fermentation; a yeast heavier than the fermenting liquid is used for beer, whereas a yeast lighter than the fermenting liquid is used for ale. In fermentation, maltose plus water produces alcohol plus carbon dioxide, and heat is released (280 Btu per pound of maltose). Some of the CO₂ is collected and later added to the finished product before pasteurization. Fermentation rooms are kept at 45 F (55 F for ale) and aging rooms at 34 F (45 F for ale). Since 1933

¹⁶ "The Beginning of the Ice Cream Industry in Baltimore." *Baltimore*, Vol. 15, No. 3 (January, 1947), p. 51.

many centrifugal compressors have been installed in new breweries or have replaced the reciprocating machines in old breweries.

Wineries require refrigeration to remove the heat of fermentation and to hasten the precipitation of cream of tartar (potassium bitartrate) in newly made wines. The fermentation room should be under 85 F. From 150,000 to 250,000 Btu is evolved per 1000 gal. The newly made wines are chilled to just above the freezing temperature, which varies from 7 to 22 F, depending upon the type of wine. Refrigeration coils may be placed in the tanks, the wine may be pumped over or through a cooler, or the room itself may be air-conditioned with refrigeration equipment

Bottled carbonated-beverage plants use refrigeration to cool the prepared sirups and to cool the water used before it is carbonated. Since a water temperature close to freezing is desired, Baudelot or descending film-type water coolers are generally used (see Fig. 11.25). The lower the water temperature, the less is the pressure required to obtain the desired carbonation.

19.9. Food Transportation. Means must be provided for preserving food products as they are carried by plane, ship, railroad, or truck from producer to distributor. The use of airplanes has been mainly confined to date to transporting fresh but highly perishable products that are locally out of season, such as asparagus and certain berries. In 1947 plans were announced for air transport of fruits and vegetables from the farmer to the city for the new method of distributing fresh produce in pre-packaged consumer units (see §19.7).

Refrigerated cargo ships have been used for years. In general, they are classified as refrigerated throughout for frozen cargo, refrigerated throughout for chilled or frozen cargo, or partially refrigerated for chilled or frozen cargo. The hulls of these vessels should be designed to be insulated properly and to carry the necessary refrigerating equipment. Most ships are designed to carry specific products over a given trade route, so that design conditions for the cooling load are known in advance. In modern ships rock or glass wool in batt form is placed against the steel-work and compressed by tongue-and-groove sheathing.

The American Bureau of Shipping requires refrigeration equipment of capacity to maintain required temperatures in tropical climates with the machines running 18 hr per day. Lloyd's rules require not less than two interchangeable complete units each of which can maintain conditions in the tropics when running 24 hr per day. Most old vessels have CO₂ equipment, but most new merchant vessels built in the United States use Freon-12 or Freon-11. Brine is generally chilled and circulated to the cargo spaces for use in gravity coils or blower units, although direct-expansion units are used in certain instances. Some recent American vessels equipped with modern dehumidification and ventilation systems are regularly shipping grapefruit from Texas to Belgium, and onions,

oranges, lemons, and apples from our west coast to the Philippines without refrigeration, and with no spoilage.

Fresh food was first shipped by *railroad cars* over 80 years ago by using ice in freight cars. Since then two types of refrigerator cars have been developed: brine tank cars and basket bunker cars. Although internal

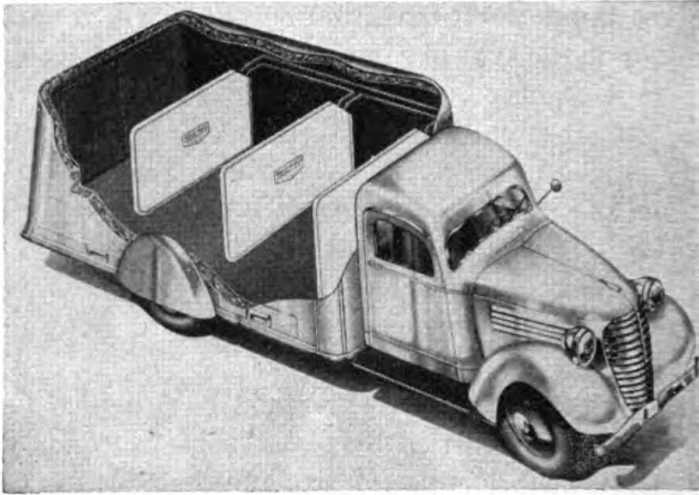


Fig. 19.3.

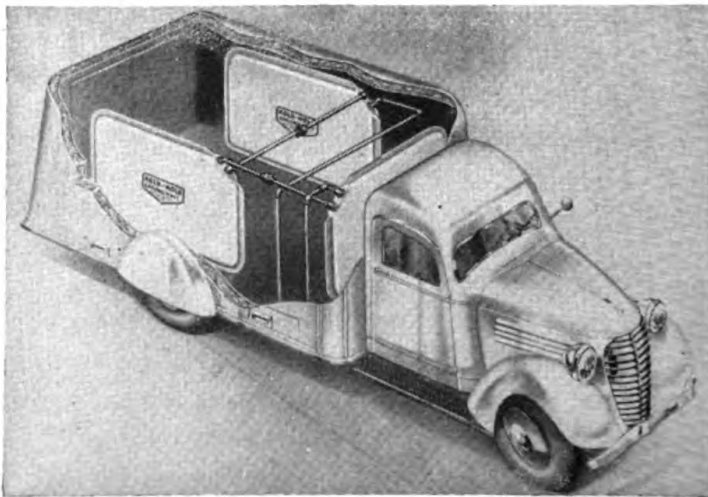


Fig. 19.4.

Methods of installing plate coolers in refrigerated trucks.
Courtesy Kold-Hold Mfg. Co.

structural details have gradually improved over the past 30 years, the general design of the car is the same, with a load space and with brine tanks or ice bunkers at each end that can be iced from the top of the car. Railroads are reluctant to change basic car designs, and considerable investments have been made in equipment including icing stations along the right of way, so that ice is still used almost exclusively. Some interest

has been created in applying mechanical refrigeration, but more developmental work is still needed in order to improve and modernize this important application of refrigeration.

The volume of food, frozen and otherwise, being shipped by *refrigerated trucks* has increased appreciably in recent years. The long-haul, trailer-type trucks use condensing units powered by a separate internal-combustion engine, by direct mechanical drive from the transmission, or by an electric motor obtaining power from a generator driven by a separate engine or driven by a power take-off from the transmission. Forced-air coolers and refrigerated plates have served as low-side equipment. Experimental work has been continued since World War II in an effort to improve the existing equipment and methods used. Heat-pump units have been tried, since heating of the space is often required for winter shipments.¹⁷

Fleets of local delivery trucks are often equipped with eutectic hold-over systems. These employ a flat tank containing an eutectic brine solution and evaporator coils. The evaporator coils are connected either to a condensing unit driven by an electric motor "plugged in" at the garage or to a central plant system when in the garage. Dry-ice systems and water ice with brine systems are also used. Methods of installing plate coolers in medium-sized trucks are shown in Figs. 19.3 and 19.4.

19.10. Commercial Refrigeration. Although the term "commercial refrigeration" has been defined in different ways, as applied here it refers to the refrigeration used in retail stores, restaurants, hotels, and institutions for foods and beverages.

Refrigerated display cases of several types as diagramed in Fig. 19.5 are used by grocers and meat dealers in order to preserve perishable merchandise as well as to display it attractively. The single-duty top display case must be placed on a base cabinet, which can be used as dry-storage space. The case is cooled by an evaporator coil at the top and by a coil under the sloping shelf. The sloping shelf and lower coil may be replaced by a refrigerated plate, or they may be omitted, with provision made for placing cracked ice in the bottom partition, on which fish and fowl may be laid directly. In a double-duty case, as shown in Fig. 19.6, the base is integral with the top and has a coil located in the front of it. The two spaces may be separated by a partition, or they may form one large space that may be partially divided by the display shelf. In the units described above, double or triple glass is used in the top portion only, and it is impossible to see into the lower portion except by opening the back bottom door. A full-vision delicatessen case is taller and has a much larger display space and window with three or more shelves for products.

¹⁷ Kirkpatrick, H. O., "Motor Transport Refrigeration." *Refrigerating Engineering*, Vol. 52, No. 6 (December, 1946), p. 521.

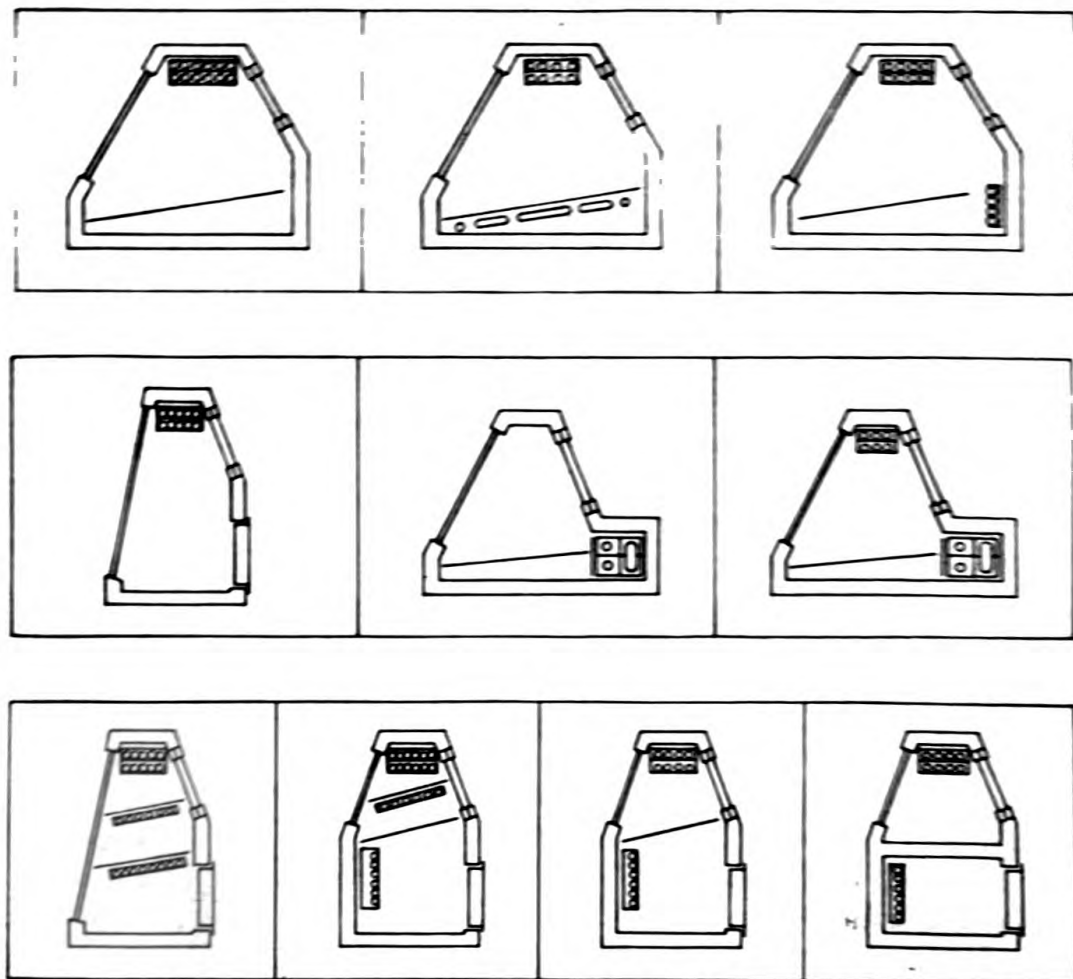


Fig. 19.5. Various types of display cases. Note the evaporator locations.

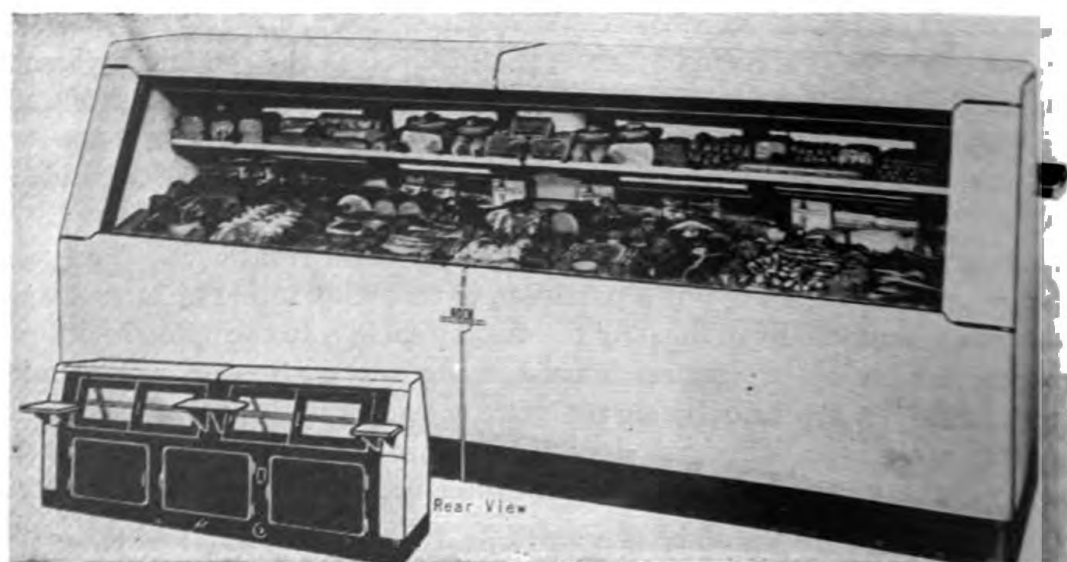


Fig. 19.6. Double-duty display case. Courtesy Koch Refrigerators.

The bottom storage compartment is much the same as for the other cases. The use of self-service display cases like those shown in Fig. 19.8 is increasing. The evaporator is located at the top rear. A glass guard in front,

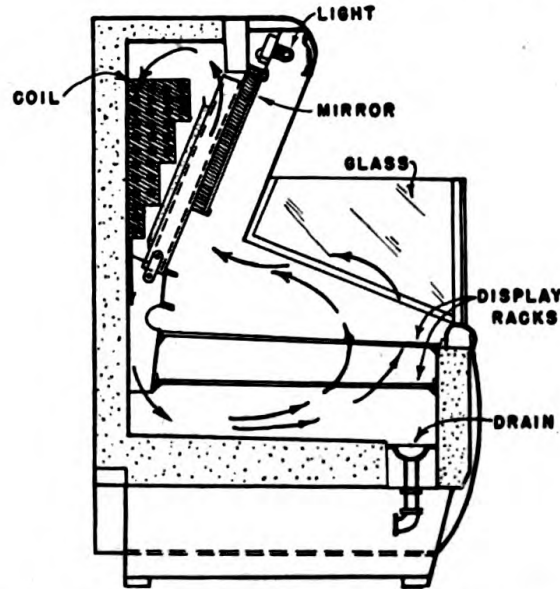


Fig. 19.7. Diagrammatic view of self-service display case.

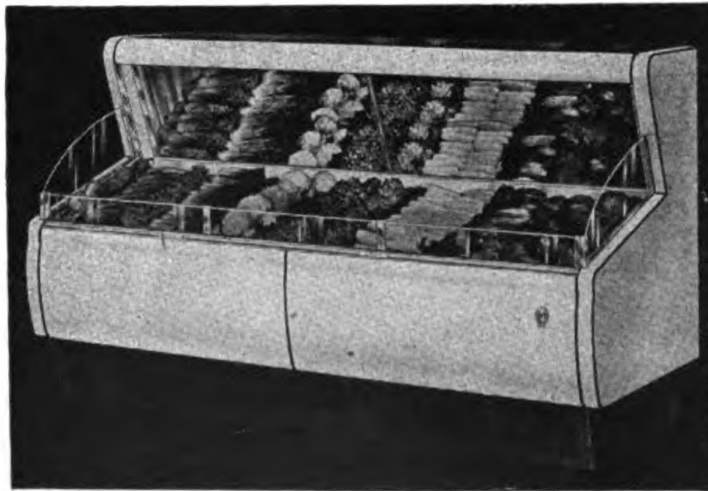


Fig. 19.8. Self-service display case. Courtesy Weber Showcase & Fixture Co., Inc.

over which the customer can reach, prevents cold air from spilling out. Reach-in, wall, and dairy refrigerators with sliding glass doors are also used in self-service markets, besides walk-in types as shown in Fig. 19.9. An air-cooled compressor of $\frac{1}{4}$ hp or larger is generally located at the bottom near one end of the above units.

Walk-in coolers of the built-in type and of the portable type shipped "knocked down" are used in the back spaces of stores and in or near

kitchens. Reach-in refrigerators of 20- to 100-cu ft capacity are also used. Many of these now employ small blower-type evaporators as shown in Fig. 11.20. Chest-type cabinets, used for ice cream, are designated by the number of holes and whether single-row or double-row.

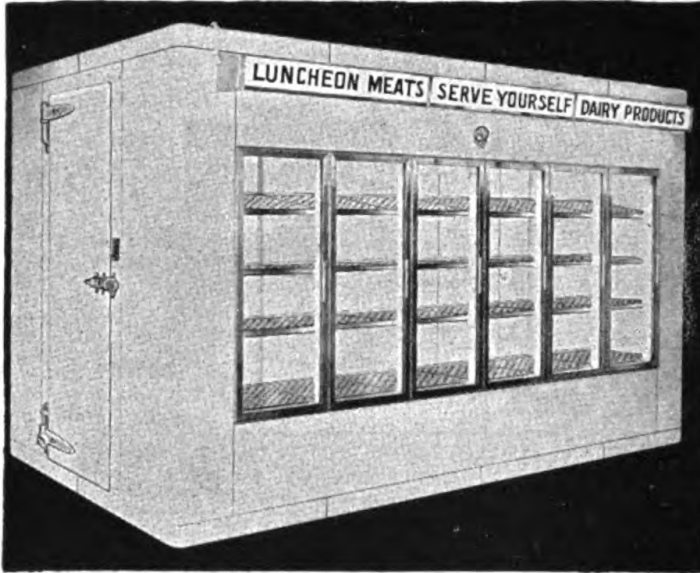


Fig. 19.9. Walk-in self-service refrigerator. Courtesy Weber Showcase & Fixture Co., Inc.

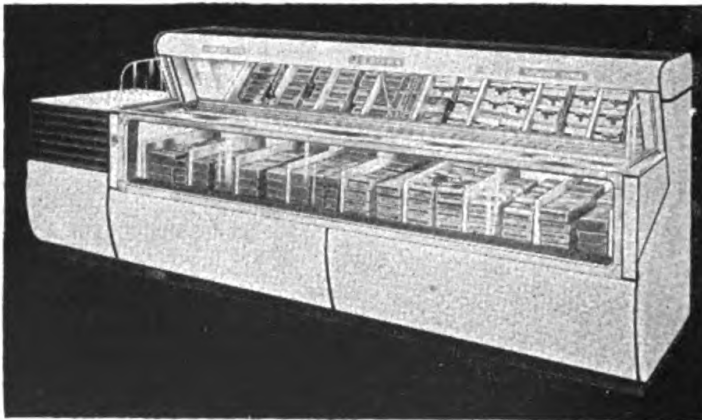


Fig. 19.10. Frozen-food display case. Courtesy Weber Showcase & Fixture Mfg. Co., Inc.

Frozen-food cabinets are of the chest type, similar to ice cream cabinets, or of the display type as shown in Fig. 19.10. Beverage coolers of the chest type are either dry or wet. The latter type contains a water tank in which the bottles are placed, and which is surrounded by the evaporator coil. A back bar refrigerator for soda fountain use is a combination unit with several types of refrigerated spaces for storing the various products. Florists' refrigerators are reach-in or walk-in units with a glass front. Many types of refrigerated beer-dispensing units have

been installed, some of which are connected with the keg storage room, and in which the line from the keg to the dispenser is also cooled.

Temperatures maintained in commercial refrigeration units vary from 34 to 50 F, depending upon the actual use. An exception are those used for frozen products, for which subzero temperatures are desired. Air-cooled compressors are generally used, so that water connections are eliminated, but good air circulation around the condenser is needed. Many of the above units, including the compressor, are factory-assembled and shipped as a package ready to plug in.

19.11. Domestic Refrigeration. Modern units for home use are of two types, freezer cabinets or boxes for food freezing or frozen-food storage only, and domestic refrigerators in which most of the space is for storage above 32 F with variable amounts of space at subfreezing temperatures.

Home and Farm Freezers. Home and farm freezers were used by a few people before 1930, but since 1938 the number of units in operation has increased rapidly. Many of the prewar units were tailor-made or else were converted ice-cream cabinets, but a great many companies started making freezers after World War II. The three general types are (1) the upright or reach-in, similar to Fig. 19.11; (2) the chest or top-opening type like the one in Fig. 19.12; and (3) the walk-in type. The first conserves floor space and has certain advantages of accessibility. The disadvantage of high door loss is overcome somewhat by inner drawers or doors, but these reduce the product storage space. Chest types are less expensive and probably more economical to operate. Wire baskets are often used to prevent misplacing and to aid in removing products. The walk-in type like that shown in Fig. 19.13 usually is entered through a small cooler kept at 32 to 40 F and which provides space for the storage of large quantities of food. Some of the larger units in all types separate the freezing and storage compartments.

Home building of units requires special care in applying the insulation and in sizing and installing the evaporator and condensing unit. Particular care must be given to vapor-sealing the insulation space and to the door or lid design.

In 1948 the units ranged from the 2.5-cu ft size selling for about \$150 to the 30 cu ft selling for \$700 to \$900, the most popular sizes being between 8 and 20 cu ft. Air-cooled compressor sizes of $\frac{1}{8}$, $\frac{1}{4}$, $\frac{1}{2}$, $\frac{3}{4}$, and 1 hp were used.

Domestic Refrigerators. Great strides in the development and improvement of these units have been made in recent years and the average retail price dropped from \$600 in 1920 to about \$150 in 1940. Annual sales for four different years before the war were over 2,000,000 units. The common sizes range from 3 to 8 cu ft, with the 6-cu ft size accounting for almost half the sales in 1940. In addition to lower costs, the trend has also been to increased storage volume, reduced energy con-

sumption, more cooling capacity, less noise, and more attractive appearance. Inside the units, high-humidity compartments for fresh vegetables and increasingly larger frozen-food compartments have been included.

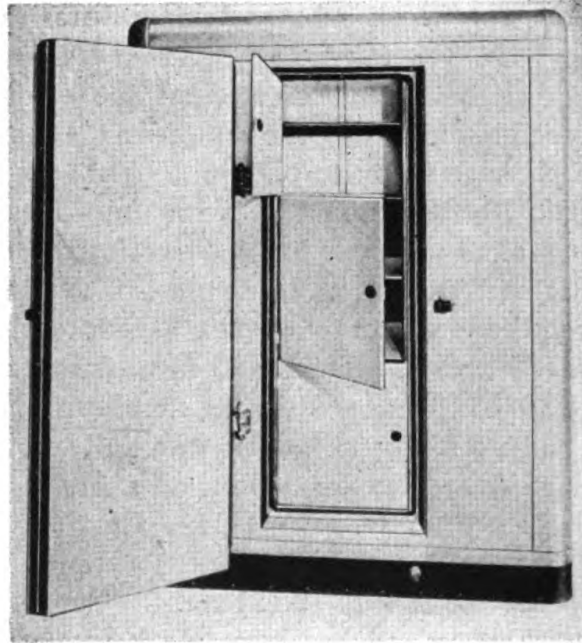


Fig. 19.11. Upright freezer. Courtesy Beall Pipe and Tank Corp.

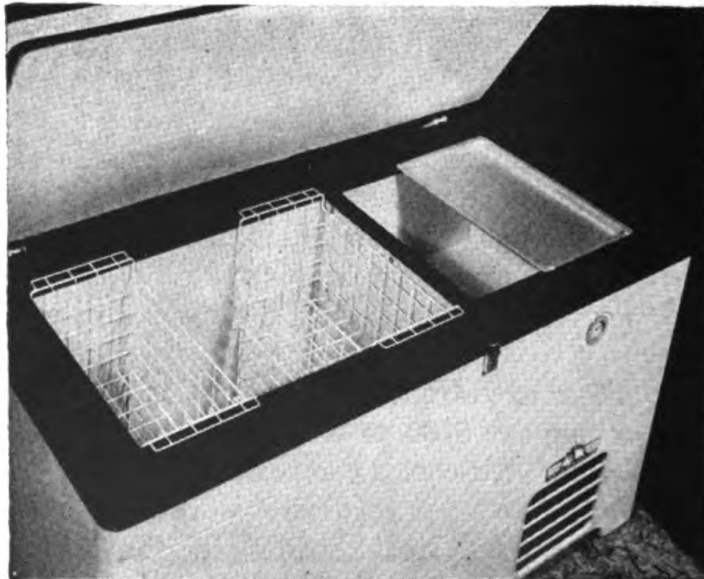


Fig. 19.12. Chest-type freezer. Courtesy Sears, Roebuck & Co.

One type of design is shown in Fig. 19.14. Anticipated postwar gadgets such as automatic ice-cube dispensing, a chilled-water faucet and revolving storage bins have been slow in materializing.

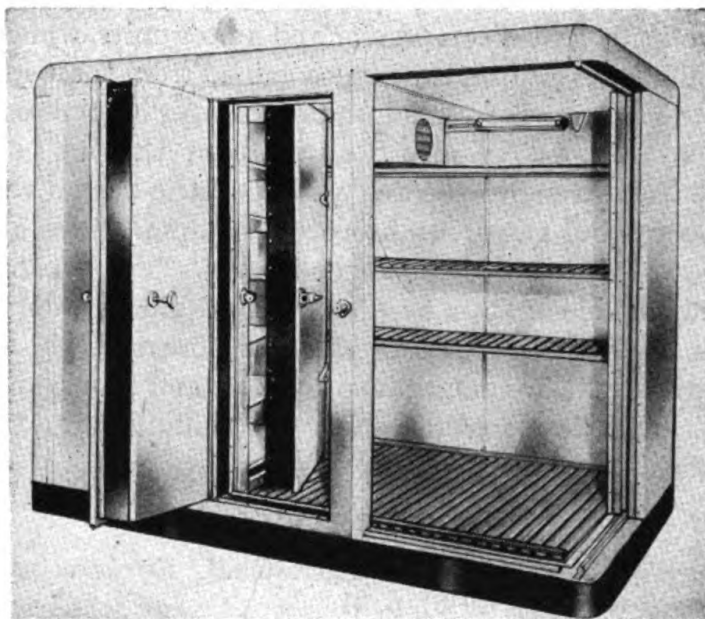


Fig. 19.13. Walk-in two-temperature freezer and storage unit—partly cut away. Courtesy Beall Pipe and Tank Corp.

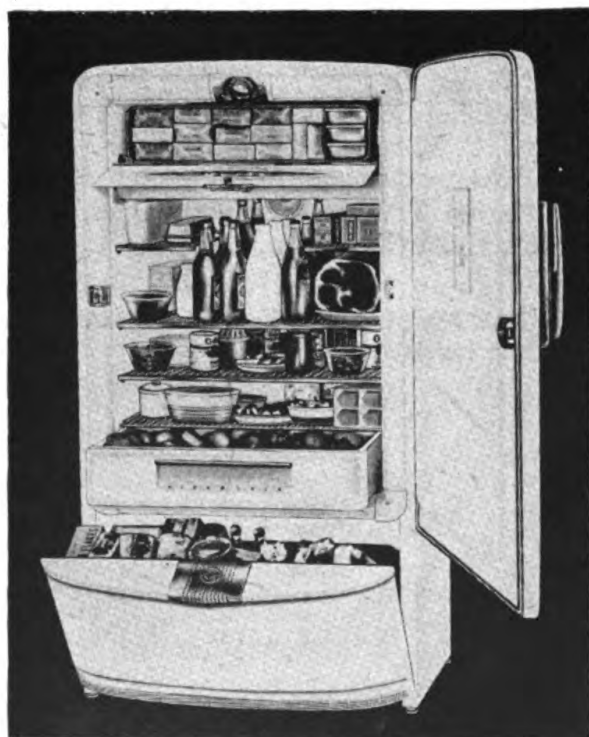


Fig. 19.14. Domestic refrigerator. Courtesy Norge Division, Borg Warner Corp.

The great improvement in the mechanical equipment was the development of hermetically sealed compressor and motor units, which eliminated shaft-seal trouble. Based on experience gained from these in the domestic field, the trend is toward sealed units for larger compressors. Reciprocating-piston compressors electrically driven through a crank and connecting rod or through a Scotch yoke, electric-driven rotary-pump-type compressors, and gas-fired absorption equipment are all used with gravity-cooled plate or finned coiled condensers. The expansion devices employed are high-side floats, a fixed orifice, or capillary tubing. Control is by a manually adjusted pressurestat that operates the compressor. Capacity requirements are determined by tests and codes, and standards of many types have been formulated by the manufacturers' associations and the technical societies involved.¹⁸

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¹⁸ *Refrigerating Data Book*, 5th ed. New York: American Society of Refrigerating Engineers, 1943, "Domestic Refrigerators," Chapter 25, p. 341.

Appendix

TABLE A.1
LETTER SYMBOLS*

a = heat transfer across an air space	p_s = saturation pressure of water vapor
A = area	p_{wb} = saturation pressure at wet-bulb temperature
c = specific heat	Pr = Prandtl number
c_p = specific heat at constant pressure	q = quantity of heat (per unit time)
c_v = specific heat at constant volume	Q = quantity of heat, total
C = thermal conductance	r = radius
c.p. = coefficient of performance	R = gas constant
d = diameter, inches	Re = Reynolds number
D = diameter, feet	s = entropy (per unit weight)
E = energy	S = entropy, total
E_k = kinetic energy	t = temperature, C or F
E_p = potential energy	t_p = triple-point temperature
f = film coefficient of heat transfer; friction coefficient	T = temperature, absolute
F = force	T.H. = total heat
g = gravitational acceleration	u = internal energy (per unit weight)
Gr = Grashof number	U = internal energy, total; over-all coefficient of heat transfer
h = enthalpy (per unit mass); coefficient of heat transfer	v = volume, specific
h_c = convection heat-transfer coefficient	V = volume, total; velocity
h_r = radiation heat-transfer coefficient	w_H = weight of water vapor per unit weight of air
$h\Sigma$ = sigma heat content	W = weight
H = enthalpy, total; head, total	W_k = work
H_R = relative humidity	x = quality of vapor; thickness
J = mechanical equivalent of heat	β = thermal coefficient of cubical expansion
$\frac{1}{J}$ = heat equivalent of work	γ = specific heat ratio, c_p/c_v
k = thermal conductivity	Δ = difference between values
l = length, inches	ϵ = emissivity coefficient
L = length, feet	η = efficiency
m = mass, $\frac{w}{g}$	ω = angular velocity
n = exponent of polytropic expansion; revolutions per unit time	ρ = density (per unit volume)
Nu = Nusselt number	τ = time
p = total pressure (force per unit area)	μ = viscosity, absolute; Joule-Thomson coefficient
p_H = partial pressure of water vapor	ϕ = per cent saturation

* To the extent practical, letter symbols used throughout the text follow the recommendations of the American Standards Association. However, the breadth of subject matter covered has necessitated some overlapping of definitions and, in a few cases where confusion has resulted, certain changes.

TABLE A.2—AMMONIA
PROPERTIES OF LIQUID AND SATURATED VAPOR†

Temp.	Pressure		Volume	Density	Enthalpy from -40 F			Entropy from -40 F	
F	Abs. lb/in. ²	Gage lb/in. ²	Vapor ft ³ /lb	Vapor lb/ft ³	Liquid Btu/lb	Vapor Btu/lb	Latent Btu/lb	Liquid Btu/lb F	Vapor Btu/lb F
<i>t</i>	<i>p</i>	<i>p_a</i>	<i>v_g</i>	<i>1/v_g</i>	<i>h_f</i>	<i>h_g</i>	<i>h_{fg}</i>	<i>s_f</i>	<i>s_g</i>
-60	5.55	*18.6	44.73	0.02235	-21.2	589.6	610.8	-0.0517	1.4769
-55	6.54	*16.6	38.38	0.02605	-15.9	591.6	607.5	-0.0386	1.4631
-50	7.67	*14.3	33.08	0.03023	-10.6	593.7	604.3	-0.0256	1.4497
-45	8.95	*11.7	28.62	0.03494	-5.3	595.6	600.9	-0.0127	1.4368
-40	10.41	*8.7	24.86	0.04022	0.0	597.6	597.6	0.0000	1.4242
-38	11.04	*7.4	23.53	.04251	2.1	598.3	596.2	.0051	.4193
-36	11.71	*6.1	22.27	.04489	4.3	599.1	594.8	.0101	.4144
-34	12.41	*4.7	21.10	.04739	6.4	599.9	593.5	.0151	.4096
-32	13.14	*3.2	20.00	.04999	8.5	600.6	592.1	.0201	.4048
-30	13.90	*1.6	18.97	0.05271	10.7	601.4	590.7	0.0250	1.4001
-28	14.71	0.0	18.00	.05555	12.8	602.1	589.3	.0300	.3955
-26	15.55	0.8	17.09	.05850	14.9	602.8	587.9	.0350	.3909
-24	16.42	1.7	16.24	.06158	17.1	603.6	586.5	.0399	.3863
-22	17.34	2.6	15.43	.06479	19.2	604.3	585.1	.0448	.3818
-20	18.30	3.6	14.68	0.06813	21.4	605.0	583.6	0.0497	1.3774
-18	19.30	4.6	13.97	.07161	23.5	605.7	582.2	.0545	.3729
-16	20.34	5.6	13.29	.07522	25.6	606.4	580.8	.0594	.3686
-14	21.43	6.7	12.66	.07898	27.8	607.1	579.3	.0642	.3643
-12	22.56	7.9	12.06	.08289	30.0	607.8	577.8	.0690	.3600
-10	23.74	9.0	11.50	0.08695	32.1	608.5	576.4	0.0738	1.3558
-8	24.97	10.3	10.97	.09117	34.3	609.2	574.9	.0786	.3516
-6	26.26	11.6	10.47	.09555	36.4	609.8	573.4	.0833	.3474
-4	27.59	12.9	9.991	.1001	38.6	610.5	571.9	.0880	.3433
-2	28.98	14.3	9.541	.1048	40.7	611.1	570.4	.0928	.3393
0	30.42	15.7	9.116	0.1097	42.9	611.8	568.9	0.0975	1.3352
2	31.92	17.2	8.714	.1148	45.1	612.4	567.3	.1022	.3312
4	33.47	18.8	8.333	.1200	47.2	613.0	565.8	.1069	.3273
6	35.09	20.4	7.971	.1254	49.4	613.6	564.2	.1115	.3234
8	36.77	22.1	7.629	.1311	51.6	614.3	562.7	.1162	.3195
10	38.51	23.8	7.304	0.1369	53.8	614.9	561.1	0.1208	1.3157
12	40.31	25.6	6.996	.1429	56.0	615.5	559.5	.1254	.3118
14	42.18	27.5	6.703	.1492	58.2	616.1	557.9	.1300	.3081
16	44.12	29.4	6.425	.1556	60.3	616.6	556.3	.1346	.3043
18	46.13	31.4	6.161	.1623	62.5	617.2	554.7	.1392	.3006
20	48.21	33.5	5.910	0.1692	64.7	617.8	553.1	0.1437	1.2969
22	50.36	35.7	5.671	.1763	66.9	618.3	551.4	.1483	.2933
24	52.59	37.9	5.443	.1837	69.1	618.9	549.8	.1528	.2897
26	54.90	40.2	5.227	.1913	71.3	619.4	548.1	.1573	.2861
28	57.28	42.6	5.021	.1992	73.5	619.9	546.4	.1618	.2825
30	59.74	45.0	4.825	0.2073	75.7	620.5	544.8	0.1663	1.2790
32	62.29	47.6	4.637	.2156	77.9	621.0	543.1	.1708	.2755
34	64.91	50.2	4.459	.2243	80.1	621.5	541.4	.1753	.2721
36	67.63	52.9	4.289	.2332	82.3	622.0	539.7	.1797	.2686
38	70.43	55.7	4.126	.2423	84.6	622.5	537.9	.1841	.2652

* Inches of mercury below one atmosphere.

† Abstracted, by permission, from "Tables of Thermodynamic Properties of Ammonia," U. S. Department of Commerce, Bureau of Standards Circular No. 142, 1945.

TABLE A.2—AMMONIA
PROPERTIES OF LIQUID AND SATURATED VAPOR (CONT.)

Temp.	Pressure		Volume	Density	Enthalpy from -40 F			Entropy from -40 F	
F	Abs. lb/in ²	Gage lb/in ²	Vapor ft ³ /lb	Vapor lb/ft ³	Liquid Btu/lb	Vapor Btu/lb	Latent Btu/lb	Liquid Btu/lb F	Vapor Btu/lb F
<i>t</i>	<i>p</i>	<i>p_g</i>	<i>v_g</i>	<i>1/v_g</i>	<i>h_f</i>	<i>h_g</i>	<i>h_{fg}</i>	<i>s_f</i>	<i>s_g</i>
40	73.32	58.6	3.971	0.2518	86.8	623.0	536.2	0.1885	1.2618
42	76.31	61.6	3.823	.2616	89.0	623.4	534.4	.1930	.2585
44	79.38	64.7	3.682	.2716	91.2	623.9	532.7	.1974	.2552
46	82.55	67.9	3.547	.2819	93.5	624.4	530.9	.2018	.2519
48	85.82	71.1	3.418	.2926	95.7	624.8	529.1	.2062	.2486
50	89.19	74.5	3.294	0.3036	97.9	625.2	527.3	0.2105	1.2453
52	92.66	78.0	3.176	.3149	100.2	625.7	525.5	.2149	.2421
54	96.23	81.5	3.063	.3265	102.4	626.1	523.7	.2192	.2389
56	99.91	85.2	2.954	.3385	104.7	626.5	521.8	.2236	.2357
58	103.7	89.0	2.851	.3508	106.9	626.9	520.0	.2279	.2325
60	107.6	92.9	2.751	0.3635	109.2	627.3	518.1	0.2322	1.2294
62	111.6	96.9	2.656	.3765	111.5	627.7	516.2	.2365	.2262
64	115.7	101.0	2.565	.3899	113.7	628.0	514.3	.2408	.2231
66	120.0	105.3	2.477	.4037	116.0	628.4	512.4	.2451	.2201
68	124.3	109.6	2.393	.4179	118.3	628.8	510.5	.2494	.2170
70	128.8	114.1	2.312	0.4325	120.5	629.1	508.6	0.2537	1.2140
72	133.4	118.7	2.235	.4474	122.8	629.4	506.6	.2579	.2110
74	138.1	123.4	2.161	.4628	125.1	629.8	504.7	.2622	.2080
76	143.0	128.3	2.089	.4786	127.4	620.1	502.7	.2664	.2050
78	147.9	133.2	2.021	.4949	129.7	630.4	500.7	.2706	.2020
80	153.0	138.3	1.955	0.5115	132.0	630.7	498.7	0.2749	1.1991
82	158.3	143.6	1.892	.5287	134.3	631.0	496.7	.2791	.1962
84	163.7	149.0	1.831	.5462	136.6	631.3	494.7	.2833	.1933
86	169.2	154.5	1.772	.5643	138.9	631.5	492.6	.2875	.1904
88	174.8	160.1	1.716	.5828	141.2	631.8	490.6	.2917	.1875
90	180.6	165.9	1.661	0.6019	143.5	632.0	488.5	0.2958	1.1846
92	186.6	171.9	1.609	.6214	145.8	632.2	486.4	.3000	.1818
94	192.7	178.0	1.559	.6415	148.2	632.5	484.3	.3041	.1789
96	198.9	184.2	1.510	.6620	150.5	632.6	482.1	.3083	.1761
98	205.3	190.6	1.464	.6832	152.9	632.9	480.0	.3125	.1733
100	211.9	197.2	1.419	0.7048	155.2	633.0	477.8	0.3166	1.1705
102	218.6	203.9	1.375	.7270	157.6	633.2	475.6	.3207	.1677
104	225.4	210.7	1.334	.7498	159.9	633.4	473.5	.3248	.1649
106	232.5	217.8	1.293	.7732	162.3	633.5	471.2	.3289	.1621
108	239.7	225.0	1.254	.7972	164.6	633.6	469.0	.3330	.1593
110	247.0	232.3	1.217	0.8219	167.0	633.7	466.7	0.3372	1.1566
112	254.5	239.8	1.180	.8471	169.4	633.8	464.4	.3413	.1538
114	262.2	247.5	1.145	.8730	171.8	633.9	462.1	.3453	.1510
116	270.1	255.4	1.112	.8996	174.2	634.0	459.8	.3495	.1483
118	278.2	263.5	1.079	.9269	176.6	634.0	457.4	.3535	.1455
120	286.4	271.7	1.047	0.9549	179.0	634.0	455.0	0.3576	1.1427
122	294.8	280.1	1.017	.9837	181.4	634.0	452.6	.3618	.1400
124	303.4	288.7	0.987	1.0132	183.9	634.0	450.1	.3659	.1372

TABLE A.3—AMMONIA
PROPERTIES OF SUPERHEATED VAPOR*

Absolute Pressure in lb/in. ² (Saturation Temperature in italics)												
Temp. F	5			10			15			20		
	-63.11			-41.34			-27.29			-16.64		
<i>t</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
<i>Sat.</i>	49.31	588.3	1.4857	25.81	597.1	1.4276	17.67	602.4	1.3938	13.50	606.2	1.3700
-50	51.05	595.2	1.5025									
-40	52.36	600.3	.5149									
-30	53.67	605.4	.5269	26.58	603.2	1.4420						
-20	54.97	610.4	.5385	27.26	608.5	.4542	18.01	606.4	1.4031			
-10	56.26	615.4	.5498	27.92	613.7	.4659	18.47	611.9	.4154	13.74	610.0	1.3784
0	57.55	620.4	1.5608	28.58	618.9	1.4773	18.92	617.2	1.4272	14.09	615.5	1.3907
10	58.84	625.4	.5716	29.24	624.0	.4884	19.37	622.5	.4386	14.44	621.0	.4025
20	60.12	630.4	.5821	29.90	629.1	.4992	19.82	627.8	.4497	14.78	626.4	.4138
30	61.41	635.4	.5925	30.55	634.2	.5097	20.26	633.0	.4604	15.11	631.7	.4248
40	62.69	640.4	.6026	31.20	639.3	.5200	20.70	638.2	.4709	15.45	637.0	.4356
50	63.96	645.5	1.6125	31.85	644.4	1.5301	21.14	643.4	1.4812	15.78	642.3	1.4460
60	65.24	650.5	.6223	32.49	649.5	.5400	21.58	648.5	.4912	16.12	647.5	.4562
70	66.51	655.5	.6319	33.14	654.6	.5497	22.01	653.7	.5011	16.45	652.8	.4662
80	67.79	660.6	.6413	33.78	659.7	.5593	22.44	658.9	.5108	16.78	658.0	.4760
90	69.06	665.6	.6506	34.42	664.8	.5687	22.88	664.0	.5203	17.10	663.2	.4856
100	70.33	670.7	1.6598	35.07	670.0	1.5779	23.31	669.2	1.5296	17.43	668.5	1.4950
110	71.60	675.8	.6689	35.71	675.1	.5870	23.74	674.4	.5388	17.76	673.7	.5042
120	72.87	680.9	.6778	36.35	680.3	.5960	24.17	679.6	.5478	18.08	678.9	.5133
130	74.14	686.1	.6865	36.99	685.4	.6049	24.60	684.8	.5567	18.41	684.2	.5223
140	75.41	691.2	.6952	37.62	690.6	.6136	25.03	690.0	.5655	18.73	689.4	.5312
150	76.68	696.4	1.7038	38.26	695.8	1.6222	25.46	695.3	1.5742	19.05	694.7	1.5399
160	77.95	701.6	.7122	38.90	701.1	.6307	25.88	700.5	.5827	19.37	700.0	.5485
170	79.21	706.8	.7206	39.54	706.3	.6391	26.31	705.8	.5911	19.70	705.3	.5569
180	80.48	712.1	.7289	40.17	711.6	.6474	26.74	711.1	.5995	20.02	710.6	.5653
190	40.81	716.9	.6556	27.16	716.4	.6077	20.34	715.9	.5736
200	41.45	722.2	1.6637	27.59	721.7	1.6158	20.66	721.2	1.5817
220	28.44	732.4	.6318	21.30	732.0	.5978
240	21.94	742.8	.6135

Temp. F	25			30			35			40		
	-7.96			-0.57			5.89			11.66		
<i>Sat.</i>	10.96	609.1	1.3515	9.236	611.6	1.3364	7.991	613.6	1.3236	7.047	615.4	1.3126
0	11.19	613.8	1.3616									
10	11.47	619.4	.3738	9.492	617.8	1.3497	8.078	616.1	1.3289			
20	11.75	625.0	.3855	9.731	623.5	.3618	8.287	622.0	.3413	7.203	620.4	1.3231
30	12.03	630.4	.3967	9.966	629.1	.3733	8.493	627.7	.3532	7.387	626.3	.3353
40	12.30	635.8	.4077	10.20	634.6	.3845	8.695	633.4	.3646	7.568	632.1	.3470
50	12.57	641.2	1.4183	10.43	640.1	1.3953	8.895	638.9	1.3756	7.746	637.8	1.3583
60	12.84	646.5	.4287	10.65	645.5	.4059	9.093	644.4	.3863	7.922	643.4	.3692
70	13.11	651.8	.4388	10.88	650.9	.4161	9.289	649.9	.3967	8.096	648.9	.3797
80	13.37	657.1	.4487	11.10	656.2	.4261	9.484	655.3	.4069	8.268	654.4	.3900
90	13.64	662.4	.4584	11.33	661.6	.4359	9.677	660.7	.4168	8.439	659.9	.4000
100	13.90	667.7	1.4679	11.55	666.9	1.4456	9.869	666.1	1.4265	8.609	665.3	1.4098
110	14.17	673.0	.4772	11.77	672.2	.4550	10.06	671.5	.4360	8.777	670.7	.4194
120	14.43	678.2	.4864	11.99	677.5	.4642	10.25	676.8	.4453	8.945	676.1	.4288
130	14.69	683.5	.4954	12.21	682.9	.4733	10.44	682.2	.4545	9.112	681.5	.4381
140	14.95	688.8	.5043	12.43	688.2	.4823	10.63	687.6	.4635	9.278	686.9	.4471
150	15.21	694.1	1.5131	12.65	693.5	1.4911	10.82	692.9	1.4724	9.444	692.3	1.4561
160	15.47	699.4	.5217	12.87	698.8	.4998	11.00	698.3	.4811	9.609	697.7	.4648
170	15.73	704.7	.5303	13.08	704.2	.5083	11.19	703.7	.4897	9.774	703.1	.4735
180	15.99	710.1	.5387	13.30	709.6	.5168	11.38	709.1	.4982	9.938	708.5	.4820
190	16.25	715.4	.5470	13.52	714.9	.5251	11.56	714.5	.5066	10.10	714.0	.4904
200	16.50	720.8	1.5552	13.73	720.3	1.5334	11.75	719.9	1.5148	10.27	719.4	1.4987
220	17.02	731.6	.5713	14.16	731.1	.5495	12.12	730.7	.5311	10.59	730.3	.5150
240	17.53	742.5	.5870	14.59	742.0	.5653	12.49	741.7	.5469	10.92	741.3	.5309
260	18.04	753.4	.6025	15.02	753.0	.5808	12.86	752.7	.5624	11.24	752.3	.5465
280	15.45	764.1	.5960	13.23	763.7	.5776	11.56	763.4	.5617
300	11.88	774.6	1.5766

* Abstracted, by permission, from "Tables of Thermodynamic Properties of Ammonia," U.S. Department of Commerce, Bureau of Standards Circular No. 142, 1945.

TABLE A.3—AMMONIA
PROPERTIES OF SUPERHEATED VAPOR (CONT.)

Absolute Pressure in lb/in.² (Saturation Temperature in italics)

Temp. F	50 <i>21.67</i>			60 <i>30.21</i>			70 <i>37.70</i>			80 <i>44.40</i>		
	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
<i>Sat.</i>	5.710	618.2	1.2939	4.805	620.5	1.2787	4.151	622.4	1.2658	3.655	624.0	1.2545
30	5.838	623.4	1.3046									
40	5.988	629.5	.3169	4.933	626.8	1.2913	4.177	623.9	1.2688			
50	6.135	635.4	1.3286	5.060	632.9	1.3035	4.290	630.4	1.2816	3.712	627.7	1.2619
60	6.280	641.2	.3399	5.184	639.0	.3152	4.401	636.6	.2937	3.812	634.3	.2745
70	6.423	646.9	.3508	5.307	644.9	.3265	4.509	642.7	.3054	3.909	640.6	.2866
80	6.564	652.6	.3613	5.428	650.7	.3373	4.615	648.7	.3166	4.005	646.7	.2981
90	6.704	658.2	.3716	5.547	656.4	.3479	4.719	654.6	.3274	4.098	652.8	.3092
100	6.843	663.7	1.3816	5.665	662.1	1.3581	4.822	660.4	1.3375	4.190	658.7	1.3199
110	6.980	669.2	.3914	5.781	667.7	.3681	4.924	666.1	.3480	4.281	664.6	.3303
120	7.117	674.7	.4009	5.897	673.3	.3778	5.025	671.8	.3579	4.371	670.4	.3404
130	7.252	680.2	.4103	6.012	678.9	.3873	5.125	677.5	.3676	4.460	676.1	.3502
140	7.387	685.7	.4195	6.126	684.4	.3966	5.224	683.1	.3770	4.548	681.8	.3598
150	7.521	691.1	1.4286	6.239	689.9	1.4058	5.323	688.7	1.3863	4.635	687.5	1.3692
160	7.655	696.6	.4374	6.352	695.5	.4148	5.420	694.3	.3954	4.722	693.2	.3784
170	7.788	702.1	.4462	6.464	701.0	.4236	5.518	699.9	.4043	4.808	698.8	.3874
180	7.921	707.5	.4548	6.576	706.5	.4323	5.615	705.5	.4131	4.893	704.4	.3963
190	8.053	713.0	.4633	6.687	712.0	.4409	5.711	711.0	.4217	4.978	710.0	.4050
200	8.185	718.5	1.4716	6.798	717.5	1.4493	5.807	716.6	1.4302	5.063	715.6	1.4136
210	8.317	724.0	.4799	6.909	723.1	.4576	5.902	722.2	.4386	5.147	721.3	.4220
220	8.448	729.4	.4880	7.019	728.6	.4658	5.998	727.7	.4469	5.231	726.9	.4304
240	8.710	740.5	.5040	7.238	739.7	.4819	6.187	738.9	.4631	5.398	738.1	.4467
260	8.970	751.6	.5197	7.457	750.9	.4976	6.376	750.1	.4789	5.565	749.4	.4626
280	9.230	762.7	1.5350	7.675	762.1	1.5130	6.563	761.4	1.4943	5.730	760.7	1.4781
300	9.489	774.0	.5500	7.892	773.3	.5281	6.750	772.7	.5095	5.894	772.1	.4933
Temp. F	90 <i>50.47</i>			100 <i>56.05</i>			120 <i>66.02</i>			140 <i>74.79</i>		
	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
<i>Sat.</i>	3.266	625.3	1.2445	2.952	626.5	1.2356	2.476	628.4	1.2201	2.132	629.9	1.2068
50												
60	3.353	631.8	1.2571	2.985	629.3	1.2409						
70	3.442	638.3	.2695	3.068	636.0	.2539	2.505	631.3	1.2255			
80	3.529	644.7	.2814	3.149	642.6	.2661	2.576	638.3	.2386	2.166	633.8	1.2140
90	3.614	650.9	.2928	3.227	649.0	.2778	2.645	645.0	.2510	2.228	640.9	.2272
100	3.698	657.0	1.3038	3.304	655.2	1.2891	2.712	651.6	1.2628	2.288	647.8	1.2396
110	3.780	663.0	.3144	3.380	661.3	.2999	2.778	658.0	.2741	2.347	654.5	.2515
120	3.862	668.9	.3247	3.454	667.3	.3104	2.842	664.2	.2850	2.404	661.1	.2628
130	3.942	674.7	.3347	3.527	673.3	.3206	2.905	670.4	.2956	2.460	667.4	.2738
140	4.021	680.5	.3444	3.600	679.2	.3305	2.967	676.5	.3058	2.515	673.7	.2843
150	4.100	686.3	1.3539	3.672	685.0	1.3401	3.029	682.5	1.3157	2.569	679.9	1.2945
160	4.178	692.0	.3633	3.743	690.8	.3495	3.089	688.4	.3254	2.622	686.0	.3045
170	4.255	697.7	.3724	3.813	696.6	.3588	3.149	694.3	.3348	2.675	692.0	.3141
180	4.332	703.4	.3813	3.883	702.3	.3678	3.209	700.2	.3441	2.727	698.0	.3236
190	4.408	709.0	.3901	3.952	708.0	.3767	3.268	706.0	.3531	2.779	704.0	.3328
200	4.484	714.7	1.3988	4.021	713.7	1.3854	3.326	711.8	1.3620	2.830	709.9	1.3418
210	4.560	720.4	.4073	4.090	719.4	.3940	3.385	717.6	.3707	2.880	715.8	.3507
220	4.635	726.0	.4157	4.158	725.1	.4024	3.442	723.4	.3793	2.931	721.6	.3594
230	4.710	731.7	.4239	4.226	730.8	.4108	3.500	729.2	.3877	2.981	727.5	.3679
240	4.785	737.3	.4321	4.294	736.5	.4190	3.557	734.9	.3960	3.030	733.3	.3763
250	4.859	743.0	1.4401	4.361	742.2	1.4271	3.614	740.7	1.4042	3.080	739.2	1.3846
260	4.933	748.7	.4481	4.428	747.9	.4350	3.671	746.5	.4123	3.129	745.0	.3928
280	5.081	760.0	.4637	4.562	759.4	.4507	3.783	758.0	.4281	3.227	756.7	.4088
300	5.228	771.5	.4789	4.695	770.8	.4660	3.895	769.6	.4435	3.323	768.3	.4243

TABLE A.3—AMMONIA
PROPERTIES OF SUPERHEATED VAPOR (CONT.)

Absolute Pressure in lb/in.² (Saturation Temperatures in italics)

Temp. F	160 <i>82.64</i>			180 <i>89.78</i>			200 <i>96.34</i>			220 <i>102.42</i>		
<i>t</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
<i>Sat.</i>	1.872	631.1	1.1952	1.667	632.0	1.1850	1.502	632.7	1.1756	1.367	633.2	1.1671
90	1.914	636.6	1.2055	1.668	632.2	1.1853						
100	1.969	643.9	1.2186	1.720	639.9	1.1992						
110	2.023	651.0	.2311	1.770	647.3	.2123	1.567	643.4	1.1947	1.400	639.4	1.1781
120	2.075	657.8	.2429	1.818	654.4	.2247	1.612	650.9	.2077	1.443	647.3	.1917
130	2.125	664.4	.2542	1.865	661.3	.2364	1.656	658.1	.2200	1.485	654.8	.2045
140	2.175	670.9	.2652	1.910	668.0	.2477	1.698	665.0	.2317	1.525	662.0	.2167
150	2.224	677.2	1.2757	1.955	674.6	1.2586	1.740	671.8	1.2429	1.564	669.0	1.2281
160	2.272	683.5	.2859	1.999	681.0	.2691	1.780	678.4	.2537	1.601	675.8	.2394
170	2.319	689.7	.2958	2.042	687.3	.2792	1.820	684.9	.2641	1.638	682.5	.2501
180	2.365	695.8	.3054	2.084	693.6	.2891	1.859	691.3	.2742	1.675	689.1	.2604
190	2.411	701.9	.3148	2.126	699.8	.2987	1.897	697.7	.2840	1.710	695.5	.2704
200	2.457	707.9	1.3240	2.167	705.9	1.3081	1.935	703.9	1.2935	1.745	701.9	1.2801
210	2.502	713.9	.3331	2.208	712.0	.3172	1.972	710.1	.3029	1.780	708.2	.2896
220	2.547	719.9	.3419	2.248	718.1	.3262	2.009	716.3	.3120	1.814	714.4	.2989
230	2.591	725.8	.3506	2.288	724.1	.3350	2.046	722.4	.3209	1.848	720.6	.3079
240	2.635	731.7	.3591	2.328	730.1	.3436	2.082	728.4	.3296	1.881	726.8	.3168
250	2.679	737.6	1.3675	2.367	736.1	1.3521	2.118	734.5	1.3382	1.914	732.9	1.3255
260	2.723	743.5	.3757	2.407	742.0	.3605	2.154	740.5	.3467	1.947	739.0	.3340
270	2.766	749.4	.3838	2.446	748.0	.3687	2.189	746.5	.3550	1.980	745.1	.3424
280	2.809	755.3	.3919	2.484	753.9	.3768	2.225	752.5	.3631	2.012	751.1	.3507
290	2.852	761.2	.3998	2.523	759.9	.3847	2.260	758.5	.3712	2.044	757.2	.3588
300	2.895	767.1	1.4076	2.561	765.8	1.3926	2.295	764.5	1.3791	2.076	763.2	1.3668
320	2.980	778.9	.4229	2.637	777.7	.4081	2.364	776.5	.3947	2.140	775.3	.3825
340	3.064	790.7	.4379	2.713	789.6	.4231	2.432	788.5	.4099	2.203	787.4	.3978
360	2.500	800.5	.4247	2.265	799.5	.4127
380	2.568	812.5	.4392	2.327	811.6	.4273
Temp. F	240 <i>108.09</i>			260 <i>113.42</i>			280 <i>118.45</i>			300 <i>123.21</i>		
<i>Sat.</i>	1.253	633.6	1.1592	1.155	633.9	1.1518	1.072	634.0	1.1449	0.999	634.0	1.1383
110	1.261	635.3	1.1621									
120	1.302	643.5	.1764	1.182	639.5	1.1617	1.078	635.4	1.1473			
130	1.342	651.3	.1898	1.220	647.8	.1757	1.115	644.0	.1621	1.023	640.1	1.1487
140	1.380	658.8	.2025	1.257	655.6	.1889	1.151	652.2	.1759	1.058	648.7	.1632
150	1.416	666.1	1.2145	1.292	663.1	1.2014	1.184	660.1	1.1888	1.091	656.9	1.1767
160	1.452	673.1	.2259	1.326	670.4	.2132	1.217	667.6	.2011	1.123	664.7	.1894
170	1.487	680.0	.2369	1.359	677.5	.2245	1.249	674.9	.2127	1.153	672.2	.2014
180	1.521	686.7	.2475	1.391	684.4	.2354	1.279	681.9	.2239	1.183	679.5	.2129
190	1.554	693.3	.2577	1.422	691.1	.2458	1.309	688.9	.2346	1.211	686.5	.2239
200	1.587	699.8	1.2677	1.453	697.7	1.2560	1.339	695.6	1.2449	1.239	693.5	1.2344
210	1.619	706.2	.2773	1.484	704.3	.2658	1.367	702.3	.2550	1.267	700.3	.2447
220	1.651	712.6	.2867	1.514	710.7	.2754	1.396	708.8	.2647	1.294	706.9	.2546
230	1.683	718.9	.2959	1.543	717.1	.2847	1.424	715.3	.2742	1.320	713.5	.2642
240	1.714	725.1	.3049	1.572	723.4	.2938	1.451	721.8	.2834	1.346	720.0	.2736
250	1.745	731.3	1.3137	1.601	729.7	1.3027	1.478	728.1	1.2924	1.372	726.5	1.2827
260	1.775	737.5	.3224	1.630	736.0	.3115	1.505	734.4	.3013	1.397	732.9	.2917
270	1.805	743.6	.3308	1.658	742.2	.3200	1.532	740.7	.3099	1.422	739.2	.3004
280	1.835	749.8	.3392	1.686	748.4	.3285	1.558	747.0	.3184	1.447	745.5	.3090
290	1.865	755.9	.3474	1.714	754.5	.3367	1.584	753.2	.3268	1.472	751.8	.3175
300	1.895	762.0	1.3554	1.741	760.7	1.3449	1.610	759.4	1.3350	1.496	758.1	1.3257
320	1.954	774.1	.3712	1.796	772.9	.3608	1.661	771.7	.3511	1.544	770.5	.3419
340	2.012	786.3	.3866	1.850	785.2	.3763	1.712	784.0	.3667	1.592	782.9	.3576
360	2.069	798.4	.4016	1.904	797.4	.3914	1.762	796.3	.3819	1.639	795.3	.3729
380	2.126	810.6	.4163	1.957	809.6	.4062	1.811	808.7	.3967	1.686	807.7	.3878
400	2.009	821.9	1.4206	1.861	821.0	1.4112	1.732	820.1	1.4024

TABLE A.4—METHYL CHLORIDE
PROPERTIES OF LIQUID AND SATURATED VAPOR†

Temp.	Pressure		Volume		Density		Enthalpy from -40 F			Entropy from -40 F	
F	Abs. lb/in. ²	Gage lb/in. ²	Liquid ft ³ /lb	Vapor ft ³ /lb	Liquid lb/ft ³	Vapor lb/ft ³	Liquid Btu/lb	Latent Btu/lb	Vapor Btu/lb	Liquid Btu/lb F	Vapor Btu/lb F
<i>t</i>	<i>p</i>	<i>p_g</i>	<i>v_f</i>	<i>v_g</i>	<i>1/v_f</i>	<i>1/v_g</i>	<i>h_f</i>	<i>h_{fg}</i>	<i>h_g</i>	<i>s_f</i>	<i>s_g</i>
-80	1.953	25.94*	0.01493	41.08	66.98	0.02434	-13.888	198.64	184.75	-0.0351	0.4882
-70	2.751	24.32*	.01508	29.84	66.31	.03351	-10.521	196.77	186.25	-0.0261	.4790
-60	3.799	22.19*	.01523	22.09	65.66	.04527	-7.039	194.78	187.74	-0.0172	.4703
-50	5.155	19.43*	.01538	16.64	65.02	.06010	-3.532	192.72	189.19	-0.0085	.4620
-40	6.878	15.92*	0.01553	12.72	64.39	0.07861	0.000	190.66	190.66	0.0000	0.4544
-38	7.272	15.12*	.01556	12.08	64.27	.08278	0.713	190.23	190.95	.0017	.4529
-36	7.684	14.28*	.01559	11.48	64.14	.08712	1.426	189.81	191.23	.0034	.4515
-34	8.115	13.40*	.01562	10.91	64.02	.09166	2.138	189.38	191.51	.0051	.4500
-32	8.566	12.48*	.01565	10.38	63.90	.09639	2.850	188.95	191.80	.0067	.4486
-30	9.036	11.52*	0.01568	9.873	63.78	0.1013	3.562	188.52	192.08	0.0084	0.4472
-28	9.526	10.53*	.01571	9.399	63.65	.1064	4.277	188.09	192.37	.0100	.4458
-26	10.04	9.490*	.01574	8.953	63.53	.1117	4.993	187.65	192.65	.0117	.4445
-24	10.57	8.399*	.01577	8.533	63.41	.1172	5.711	187.22	192.93	.0133	.4431
-22	11.13	7.267*	.01580	8.136	63.29	.1229	6.427	186.78	193.21	.0150	.4418
-20	11.71	6.090*	0.01583	7.761	63.17	0.1289	7.146	186.34	193.49	0.0166	0.4405
-18	12.31	4.866*	.01586	7.408	63.86	.1350	7.863	185.90	193.76	.0183	.4393
-16	12.93	3.594*	.01589	7.074	62.93	.1414	8.584	185.46	194.04	.0199	.4380
-14	13.58	2.268*	.01592	6.758	62.81	.1480	9.307	185.01	194.32	.0215	.4367
-12	14.26	0.890*	.01595	6.459	62.70	.1548	10.03	184.56	194.59	.0232	.4355
-10	14.96	0.266	0.01598	6.176	62.58	0.1619	10.75	184.11	194.87	0.0247	0.4343
-8	15.69	0.996	.01601	5.908	62.46	.1693	11.48	183.66	195.14	.0263	.4331
-6	16.45	1.754	.01604	5.654	62.34	.1769	12.20	183.21	195.42	.0279	.4319
-4	17.24	2.540	.01607	5.413	62.23	.1847	12.93	182.76	195.69	.0295	.4307
-2	18.05	3.356	.01610	5.185	62.11	.1929	13.66	182.30	195.96	.0311	.4296
0	18.90	4.201	0.01613	4.969	62.00	0.2013	14.39	181.85	196.23	0.0327	0.4284
2	19.77	5.077	.01616	4.763	61.88	.2100	15.12	181.39	196.51	.0343	.4273
4	20.68	5.985	.01619	4.568	61.77	.2189	15.85	180.93	196.78	.0359	.4262
5	21.15	6.455	.01622	4.471	61.65	.2237	16.21	180.70	196.92	.0367	.4257
6	21.62	6.924	.01625	4.379	61.54	.2284	16.58	180.47	197.05	.0375	.4251
8	22.59	7.896	.01628	4.206	61.43	.2378	17.31	180.01	197.31	.0390	.4240
10	23.60	8.903	0.01631	4.038	61.31	0.2477	18.04	179.53	197.58	0.0406	0.4229
12	24.64	9.943	.01634	3.878	61.20	.2579	18.77	179.06	197.83	.0422	.4218
14	25.72	11.02	.01637	3.726	61.09	.2684	19.51	178.58	198.09	.0437	.4208
16	26.83	12.13	.01640	3.581	60.98	.2792	20.25	178.10	198.34	.0453	.4198
18	27.97	13.28	.01644	3.443	60.83	.2904	20.98	177.61	198.59	.0468	.4187
20	29.16	14.46	0.01647	3.312	60.72	0.3019	21.73	177.11	198.84	0.0484	0.4177
22	30.38	15.69	.01650	3.186	60.61	.3138	22.47	176.61	199.08	.0499	.4166
24	31.64	16.95	.01654	3.067	60.46	.3261	23.21	176.11	199.32	.0514	.4156
26	32.95	18.25	.01658	2.952	60.31	.3388	23.95	175.61	199.56	.0530	.4146
28	34.29	19.60	.01662	2.843	60.17	.3517	24.70	175.10	199.79	.0545	.4136
30	35.68	20.98	0.01665	2.739	60.06	0.3650	25.44	174.59	200.03	0.0560	0.4126
32	37.11	22.41	.01669	2.640	59.92	.3787	26.18	174.08	200.26	.0575	.4117
34	38.58	23.88	.01673	2.546	59.77	.3928	26.93	173.56	200.49	.0590	.4107
36	40.09	25.39	.01677	2.455	59.63	.4073	27.67	173.05	200.72	.0605	.4098
38	41.65	26.95	.01681	2.369	59.49	.4222	28.42	172.53	200.95	.0621	.4089
40	43.25	28.56	0.01684	2.286	59.38	0.4375	29.17	172.00	201.17	0.0636	0.4079
42	44.91	30.21	.01688	2.206	59.24	.4532	29.92	171.48	201.40	.0651	.4070
44	46.61	31.91	.01692	2.130	59.10	.4694	30.67	170.95	201.62	.0665	.4061
46	48.35	33.66	.01696	2.057	58.96	.4861	31.42	170.42	201.84	.0680	.4052
48	50.15	35.45	.01700	1.987	58.82	.5033	32.17	169.89	202.06	.0695	.4043

* Inches of mercury below one atmosphere.

† Courtesy E. I. DuPont de Nemours & Co., Inc., The Electrochemicals Department.

TABLE A.4—METHYL CHLORIDE
PROPERTIES OF LIQUID AND SATURATED VAPOR (CONT.)

Temp.	Pressure		Volume		Density		Enthalpy from -40 F			Entropy from -40 F	
F	Abs. lb/in. ²	Gage lb/in. ²	Liquid ft ³ /lb	Vapor ft ³ /lb	Liquid lb/ft ³	Vapor lb/ft ³	Liquid Btu/lb	Latent Btu/lb	Vapor Btu/lb	Liquid Btu/lb F	Vapor Btu/lb F
<i>t</i>	<i>p</i>	<i>p_g</i>	<i>v_f</i>	<i>v_g</i>	<i>1/v_f</i>	<i>1/v_g</i>	<i>h_f</i>	<i>h_{fg}</i>	<i>h_g</i>	<i>s_f</i>	<i>s_g</i>
50	51.99	37.29	0.01704	1.920	58.69	0.5208	32.93	169.35	202.28	0.0710	0.4034
52	53.88	39.18	0.01708	1.856	58.55	0.5388	33.68	168.81	202.49	0.0725	0.4025
54	55.83	41.13	0.01712	1.794	58.41	0.5573	34.44	168.27	202.71	0.0740	0.4017
56	57.83	43.13	0.01716	1.735	58.28	0.5763	35.19	167.72	202.91	0.0754	0.4008
58	59.88	45.19	0.01720	1.679	58.14	0.5958	35.95	167.18	203.13	0.0769	0.3999
60	62.00	47.30	0.01724	1.624	58.00	0.6158	36.71	166.62	203.33	0.0784	0.3991
62	64.17	49.47	0.01728	1.572	57.87	0.6362	37.47	166.07	203.54	0.0798	0.3983
64	66.39	51.70	0.01732	1.522	57.74	0.6572	38.23	165.51	203.74	0.0813	0.3974
66	68.67	53.98	0.01736	1.473	57.60	0.6788	39.00	164.95	203.95	0.0827	0.3966
68	71.01	56.32	0.01740	1.427	57.47	0.7008	39.76	164.39	204.15	0.0842	0.3958
70	73.41	58.71	0.01744	1.382	57.34	0.7234	40.52	163.82	204.34	0.0856	0.3950
72	75.86	61.17	0.01748	1.339	57.21	0.7467	41.29	163.24	204.53	0.0870	0.3941
74	78.37	63.68	0.01752	1.298	57.08	0.7704	42.06	162.66	204.72	0.0885	0.3933
76	80.94	66.25	0.01756	1.258	56.95	0.7948	42.82	162.08	204.90	0.0899	0.3925
78	83.57	68.87	0.01760	1.220	56.82	0.8196	43.59	161.50	205.09	0.0913	0.3918
80	86.26	71.56	0.01764	1.183	56.69	0.8451	44.36	160.91	205.27	0.0928	0.3910
82	89.01	74.31	0.01768	1.148	56.56	0.8710	45.13	160.32	205.45	0.0942	0.3902
84	91.82	77.13	0.01773	1.114	56.40	0.8979	45.90	159.72	205.62	0.0956	0.3894
86	94.70	80.00	0.01778	1.081	56.24	0.9253	46.67	159.13	205.80	0.0970	0.3887
88	97.64	82.94	0.01782	1.049	56.12	0.9531	47.44	158.52	205.96	0.0984	0.3879
90	100.6	85.95	0.01786	1.018	55.99	0.9819	48.21	157.92	206.13	0.0998	0.3872
92	103.7	89.02	0.01791	0.9889	55.83	1.011	48.99	157.31	206.30	0.1012	0.3865
94	106.9	92.16	0.01796	0.9603	55.68	1.041	49.77	156.69	206.46	0.1026	0.3857
96	110.1	95.37	0.01800	0.9333	55.56	1.072	50.54	156.08	206.62	0.1041	0.3850
98	113.4	98.65	0.01804	0.9069	55.43	1.103	51.32	155.46	206.78	0.1055	0.3843
100	116.7	102.0	0.01808	0.8814	55.31	1.135	52.09	154.85	206.94	0.1069	0.3836
102	120.1	105.4	0.01813	0.8568	55.15	1.167	52.87	154.22	207.09	0.1082	0.3828
104	123.6	108.9	0.01818	0.8331	55.01	1.200	53.65	153.60	207.25	0.1096	0.3822
106	127.2	112.5	0.01823	0.8105	54.85	1.234	54.43	152.97	207.40	0.1110	0.3815
108	130.8	116.1	0.01828	0.7884	54.70	1.268	55.22	152.33	207.55	0.1124	0.3808
110	134.5	119.8	0.01833	0.7672	54.55	1.303	56.00	151.70	207.70	0.1138	0.3801
112	138.3	123.6	0.01838	0.7466	54.41	1.339	56.78	151.06	207.84	0.1151	0.3794
114	142.2	127.5	0.01843	0.7268	54.26	1.376	57.57	150.41	207.98	0.1165	0.3787
116	146.1	131.4	0.01848	0.7075	54.11	1.414	58.36	149.77	208.13	0.1179	0.3781
118	150.1	135.4	0.01853	0.6889	53.97	1.452	59.15	149.11	208.26	0.1193	0.3774
120	154.2	139.5	0.01859	0.6710	53.79	1.490	59.93	148.46	208.39	0.1206	0.3768
122	158.4	143.7	0.01865	0.6534	53.62	1.530	60.73	147.80	208.53	0.1220	0.3762
124	162.6	147.9	0.01870	0.6367	53.48	1.571	61.51	147.14	208.65	0.1234	0.3755
126	167.0	152.3	0.00875	0.6201	53.33	1.613	62.31	146.47	208.78	0.1247	0.3749
128	171.4	156.7	0.01881	0.6043	53.16	1.655	63.10	145.80	208.90	0.1261	0.3742
130	175.9	161.1	0.01887	0.5889	52.99	1.698	63.89	145.13	209.02	0.1274	0.3736
132	180.4	165.7	0.01893	0.5741	52.83	1.742	64.69	144.45	209.14	0.1288	0.3730
134	185.1	170.4	0.01898	0.5596	52.69	1.787	65.48	143.77	209.25	0.1301	0.3723
136	189.8	175.1	0.01904	0.5455	52.52	1.833	66.28	143.09	209.37	0.1314	0.3717
138	194.7	180.0	0.01909	0.5320	52.38	1.880	67.08	142.40	209.48	0.1328	0.3711
140	199.6	184.9	0.01915	0.5189	52.22	1.927	67.87	141.71	209.58	0.1341	0.3705
150	225.4	210.7	0.01945	0.4586	51.41	2.181	71.87	138.23	210.10	0.1407	0.3674
160	253.5	238.8	0.01978	0.4070	50.56	2.457	75.90	134.66	210.56	0.1473	0.3646
170	283.9	269.2	0.02015	0.3613	49.63	2.768	79.97	130.96	210.93	0.1538	0.3618

TABLE A.5—METHYL CHLORIDE
PROPERTIES OF SUPERHEATED VAPOR*

Temp. F.	Abs. Pressure, 6 lbs./in. ² Gage Pressure, 17.7 in. Vac. (Sat. Temp., -44.8 F.)			Abs. Pressure, 8 lbs./in. ² Gage Pressure, 13.6 in. Vac. (Sat. Temp., -34.5 F.)			Abs. Pressure, 10 lbs./in. ² Gage Pressure, 9.6 in. Vac. (Sat. Temp., -26.1 F.)			Abs. Pressure, 12 lbs./in. ² Gage Pressure, 5.6 in. Vac. (Sat. Temp., -19.0 F.)		
	<i>t</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
Sat.	14.46	189.98	.4580	11.06	191.44	.4504	8.993	192.64	.4446	7.687	193.64	.4389
-40	14.62	190.77	.4599	11.17	192.17	.4523	9.124	193.67	.4471	7.752	195.21	.4436
-30	14.99	192.52	.4640	11.46	193.97	.4563	9.346	195.50	.4512			
-20	15.36	194.27	.4681	11.74	195.78	.4603						
-10	15.72	196.06	.4721									
0	16.09	197.84	.4760	12.01	197.58	.4643	9.567	197.32	.4552	7.943	197.06	.4476
10	16.45	199.66	.4799	12.29	199.42	.4682	9.788	199.18	.4591	8.131	198.94	.4516
20	16.82	201.48	.4838	12.57	201.26	.4721	10.01	201.04	.4630	8.318	200.82	.4555
30	17.18	203.34	.4876	12.84	203.12	.4760	10.23	202.91	.4669	8.503	202.70	.4594
40	17.55	205.19	.4914	13.11	204.97	.4798	10.45	204.78	.4707	8.688	204.58	.4632
50	17.91	207.10	.4952	13.39	206.90	.4836	10.67	206.70	.4745	8.873	206.51	.4670
60	18.27	209.01	.4989	13.66	208.82	.4873	10.89	208.62	.4782	9.058	208.43	.4708
65	18.45	209.98	.5007	13.79	209.79	.4891	11.00	209.60	.4800	9.147	209.41	.4726
70	18.63	210.95	.5025	13.93	210.76	.4910	11.11	210.58	.4819	9.239	210.39	.4745
80	18.99	212.88	.5061	14.21	212.70	.4946	11.33	212.53	.4856	9.422	212.35	.4782
90	19.35	214.85	.5097	14.48	214.68	.4982	11.55	214.51	.4892	9.605	214.34	.4818
100	19.71	216.82	.5133	14.75	216.66	.5018	11.77	216.50	.4928	9.788	216.34	.4855
110	20.07	218.83	.5169	15.02	218.68	.5054	11.99	218.52	.4964	9.971	218.37	.4890
120	20.42	220.84	.5204	15.29	220.70	.5089	12.21	220.54	.5000	10.15	220.40	.4926
130	20.78	222.89	.5239	15.56	222.75	.5124	12.43	222.61	.5035	10.34	222.47	.4961
140	21.14	224.94	.5274	15.83	224.81	.5159	12.65	224.67	.5069	10.52	224.54	.4996
150	21.50	227.03	.5308	16.10	226.90	.5193	12.86	226.77	.5104	10.70	226.64	.5030
160	21.86	229.11	.5342	16.37	228.99	.5227	13.08	228.86	.5138	10.88	228.74	.5065
170	22.21	231.24	.5376	16.64	231.12	.5261	13.30	231.00	.5172	11.06	230.88	.5099
180	22.57	233.36	.5410	16.91	233.25	.5295	13.52	233.13	.5206	11.24	233.02	.5133
190	22.93	235.52	.5443	17.18	235.41	.5329	13.74	235.30	.5240	11.42	235.19	.5166
200	23.29	237.69	.5476	17.45	237.58	.5362	13.95	237.47	.5273	11.60	237.37	.5200
210	23.64	239.89	.5510	17.72	239.78	.5395	14.17	239.68	.5306	11.78	239.58	.5233
220	24.00	242.09	.5542	17.99	241.99	.5428	14.38	241.89	.5337	11.96	241.79	.5266
230	24.35	244.34	.5575	18.26	244.25	.5460	14.60	244.15	.5372	12.15	244.06	.5299
240	24.71	246.60	.5607	18.52	246.51	.5493	14.81	246.42	.5405	12.33	246.33	.5332
250	25.06	248.88	.5640	18.79	248.79	.5525	15.03	248.70	.5437	12.51	248.61	.5364
260	25.42	251.16	.5672	19.06	251.06	.5559	15.24	250.98	.5469	12.69	250.90	.5396
270	19.38	253.40	.5589	15.46	253.32	.5501	12.87	253.24	.5428
280	15.67	255.66	.5532	13.06	255.38	.5460

APPENDIX

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Temp. F.	Abs. Pressure, 14 lbs./in. ² Gage Pressure, 1.4 in. V.ao. (Sat. Temp., -12.8 F.)			Abs. Pressure, 16 lbs./in. ² Gage Pressure, 1.3 lbs./in. ² (Sat. Temp., -7.2 F.)			Abs. Pressure, 20 lbs./in. ² Gage Pressure, 5.3 lbs./in. ² (Sat. Temp., 2.6 F.)			Abs. Pressure, 24 lbs./in. ² Gage Pressure, 9.3 lbs./in. ² (Sat. Temp., 10.8 F.)		
Sat.	6.570	194.51	.4569	5.804	195.88	.4598	4.710	196.58	.4670	3.973	197.67	.4724
-10	6.614	194.92	.4371									
0	6.777	196.80	.4412	5.907	196.54	.4355	4.801	197.95	.4300	4.063	199.41	.4262
10	6.940	198.70	.4452	6.050	198.45	.4396	4.917	199.90	.4341			
20	7.102	200.60	.4491	6.192	200.37	.4435	5.032	201.82	.4380	4.161	201.35	.4301
30	7.262	202.49	.4530	6.333	202.27	.4474	5.146	203.75	.4420	4.259	203.30	.4341
40	7.422	204.38	.4569	6.473	204.18	.4513						
50	7.581	206.31	.4607	6.613	206.10	.4552	5.260	205.71	.4458	4.356	205.29	.4380
60	7.740	208.24	.4645	6.753	208.05	.4589	5.373	207.66	.4496	4.452	207.28	.4419
65	7.819	209.22	.4663	6.823	209.04	.4608	5.429	208.66	.4515	4.499	208.28	.4438
70	7.899	210.21	.4682	6.893	210.03	.4627	5.486	209.66	.4534	4.547	209.29	.4457
80	8.057	212.18	.4719	7.033	212.00	.4664	5.599	211.65	.4572	4.642	211.29	.4495
90	8.215	214.18	.4756	7.173	214.01	.4701	5.711	213.67	.4608	4.737	213.33	.4532
100	8.373	216.19	.4792	7.311	216.01	.4737	5.823	215.69	.4645	4.831	215.37	.4569
110	8.531	218.21	.4828	7.449	218.06	.4773	5.935	217.75	.4681	4.925	217.44	.4606
120	8.688	220.25	.4863	7.587	220.10	.4809	6.046	219.80	.4717	5.019	219.50	.4642
130	8.845	222.32	.4899	7.725	222.18	.4844	6.157	221.90	.4753	5.112	221.61	.4678
140	9.002	224.40	.4934	7.863	224.26	.4879	6.268	223.99	.4788	5.205	223.71	.4713
150	9.159	226.51	.4968	8.001	226.38	.4914	6.379	226.12	.4823	5.298	225.85	.4748
160	9.314	228.61	.5003	8.137	228.49	.4949	6.489	228.24	.4858	5.390	227.99	.4782
170	9.469	230.76	.5037	8.273	230.64	.4983	6.599	230.40	.4892	5.482	230.16	.4818
180	9.624	232.90	.5071	8.409	232.79	.5017	6.709	232.56	.4927	5.574	232.32	.4852
190	9.779	235.08	.5105	8.545	234.97	.5051	6.819	234.75	.4961	5.666	234.53	.4886
200	9.934	237.26	.5138	8.681	237.15	.5084	6.929	236.94	.4994	5.758	236.73	.4920
210	10.09	239.48	.5172	8.817	239.38	.5118	7.038	239.17	.5028	5.849	238.97	.4954
220	10.24	241.69	.5205	8.953	241.60	.5151	7.147	241.40	.5061	5.940	241.20	.4987
230	10.40	243.97	.5237	9.089	243.87	.5184	7.256	243.68	.5094	6.031	243.49	.5020
240	10.55	246.24	.5270	9.225	246.14	.5216	7.365	245.96	.5127	6.122	245.78	.5053
250	10.71	248.52	.5302	9.361	248.43	.5249	7.474	248.26	.5159	6.213	248.08	.5086
260	10.86	250.81	.5334	9.497	250.73	.5281	7.583	250.55	.5192	6.304	250.39	.5118
270	11.01	253.15	.5366	9.632	253.07	.5313	7.692	252.89	.5224	6.395	252.72	.5150
280	11.17	255.49	.5398	9.767	255.40	.5345	7.801	255.23	.5256	6.486	255.06	.5182
290	11.32	257.83	.5430	9.902	257.76	.5377	7.910	257.60	.5287	6.577	257.43	.5214
300	10.04	260.12	.5409	8.019	259.96	.5319	6.668	259.80	.5245
310	8.128	262.36	.5350	6.759	262.21	.5277

TABLE A.5—METHYL CHLORIDE
PROPERTIES OF SUPERHEATED VAPOR (CONT.)

Temp. F.	Abs. Pressure, 28 lbs./in. ² Gage Pressure, 13.3 lbs./in. ² (Sat. Temp., 18.0 F.)			Abs. Pressure, 32 lbs./in. ² Gage Pressure, 17.3 lbs./in. ² (Sat. Temp., 24.6 F.)			Abs. Pressure, 36 lbs./in. ² Gage Pressure, 21.3 lbs./in. ² (Sat. Temp., 30.5 F.)			Abs. Pressure, 40 lbs./in. ² Gage Pressure, 25.3 lbs./in. ² (Sat. Temp., 35.9 F.)		
t	v	h	s	v	h	s	v	h	s	v	h	s
Sat.	3.439	198.59	.4186	3.034	199.59	.4153	2.717	200.10	.4123	2.461	200.75	.4097
20	3.455	198.89	.4194	3.075	200.38	.4176	2.781	201.89	.4162	2.485	201.38	.4115
30	3.541	200.87	.4235	3.151	202.38	.4216						
40	3.626	202.84	.4275									
50	3.710	204.87	.4314	3.226	204.44	.4256						
60	3.794	206.90	.4353	3.300	206.50	.4295	2.849	203.99	.4203	2.547	203.54	.4156
65	3.835	207.91	.4372	3.337	207.52	.4314	2.916	206.10	.4243	2.608	205.69	.4196
70	3.877	208.92	.4392	3.374	208.54	.4334	2.949	207.13	.4262	2.638	206.73	.4216
80	3.960	210.95	.4430	3.447	210.58	.4372	2.982	208.16	.4282	2.668	207.77	.4236
90	4.042	212.99	.4467	3.520	212.64	.4410	3.048	210.21	.4321	2.729	209.84	.4275
100	4.124	215.04	.4504	3.592	214.70	.4447	3.113	212.29	.4359	2.788	211.94	.4313
110	4.206	217.12	.4541	3.664	216.81	.4485	3.178	214.37	.4397	2.847	214.03	.4351
120	4.287	219.21	.4578	3.736	218.91	.4521	3.243	216.49	.4434	2.906	216.17	.4389
130	4.368	221.32	.4614	3.807	221.03	.4558	3.307	218.61	.4471	2.964	218.32	.4426
140	4.449	223.44	.4649	3.878	223.16	.4593	3.371	220.75	.4508	3.022	220.46	.4463
150	4.529	225.59	.4685	3.949	225.32	.4629	3.435	222.88	.4544	3.080	222.60	.4499
160	4.609	227.73	.4720	4.020	227.48	.4664	3.499	225.05	.4579	3.138	224.78	.4535
170	4.689	229.91	.4755	4.090	229.67	.4699	3.562	227.22	.4615	3.195	226.96	.4570
180	4.768	232.09	.4789	4.160	231.86	.4734	3.625	229.42	.4650	3.252	229.18	.4606
190	4.847	234.30	.4823	4.230	234.08	.4768	3.688	231.62	.4685	3.309	231.38	.4641
200	4.926	236.51	.4857	4.300	236.30	.4802	3.751	233.85	.4719	3.366	233.62	.4675
210	5.005	238.76	.4891	4.370	238.55	.4836	3.813	236.08	.4754	3.423	235.86	.4710
220	5.084	241.00	.4924	4.439	240.80	.4870	3.875	238.34	.4788	3.479	238.13	.4744
230	5.163	243.30	.4957	4.508	243.11	.4903	3.937	240.60	.4821	3.535	240.40	.4778
240	5.241	245.59	.4990	4.577	245.41	.4936	3.999	242.92	.4855	3.591	242.72	.4811
250	5.319	247.90	.5023	4.646	247.73	.4969	4.061	245.23	.4888	3.647	245.04	.4845
260	5.397	250.21	.5056	4.715	250.04	.5002	4.122	247.55	.4921	3.703	247.37	.4878
270	5.475	252.56	.5088	4.784	252.39	.5034	4.183	249.87	.4953	3.759	249.70	.4910
280	5.553	254.90	.5120	4.852	254.74	.5066	4.244	252.23	.4986	3.814	252.06	.4943
290	5.631	257.28	.5152	4.920	257.13	.5098	4.305	254.58	.5018	3.869	254.42	.4975
300	5.709	259.66	.5184	4.988	259.51	.5130	4.366	256.97	.5050	3.924	256.82	.5007
310	5.787	262.07	.5215	5.056	261.92	.5161	4.427	259.36	.5082	3.979	259.21	.5039
320	5.865	264.48	.5246	5.124	264.34	.5192	4.488	261.78	.5114	4.034	261.64	.5071
330	5.192	266.79	.5223	4.549	264.20	.5145	4.089	264.06	.5102
340	4.610	266.67	.5176	4.144	266.53	.5133
				4.671	269.14	.5207	4.199	269.00	.5164

APPENDIX

Temp. F.	Abs. Pressure, 50 lbs./in. ² Gage Pressure, 35.3 lbs./in. ² (Sat. Temp., 47.8 F.)			Abs. Pressure, 60 lbs./in. ² Gage Pressure, 45.3 lbs./in. ² (Sat. Temp., 58.1 F.)			Abs. Pressure, 70 lbs./in. ² Gage Pressure, 55.3 lbs./in. ² (Sat. Temp., 67.1 F.)			Abs. Pressure, 80 lbs./in. ² Gage Pressure, 65.3 lbs./in. ² (Sat. Temp., 75.3 F.)		
Sat.	1.992	202.09	4043	1.676	203.17	3998	1.448	204.08	3960	1.273	204.84	3927
50	2.003	202.55	4053	1.684	203.49	4008						
60	2.054	204.65	4094	1.705	204.60	4028						
65	2.079	205.71	4114	1.727	205.71	4049						
70	2.104	206.77	4134	1.770	207.92	4089	1.458	204.72	3974	1.289	205.85	3949
80	2.154	208.89	4174	1.812	210.11	4129	1.496	206.93	4015	1.323	208.17	3991
90	2.203	211.03	4213				1.533	209.17	4056			
100	2.252	213.18	4252	1.854	212.30	4169	1.570	211.41	4096	1.356	210.50	4032
110	2.300	215.35	4290	1.895	214.50	4208	1.606	213.64	4136	1.389	212.76	4073
120	2.348	217.52	4328	1.936	216.69	4246	1.642	215.86	4175	1.421	215.02	4113
130	2.396	219.70	4366	1.977	218.93	4284	1.678	218.14	4214	1.453	217.35	4152
140	2.443	221.88	4402	2.017	221.16	4322	1.713	220.43	4253	1.485	219.68	4191
150	2.490	224.10	4439	2.057	223.41	4359	1.748	222.71	4290	1.516	222.00	4229
160	2.537	226.32	4475	2.097	225.66	4396	1.783	224.99	4327	1.547	224.32	4267
170	2.584	228.55	4511	2.137	227.92	4432	1.818	227.28	4364	1.578	226.64	4304
180	2.630	230.79	4546	2.176	230.19	4468	1.852	229.57	4400	1.608	228.96	4341
190	2.676	233.05	4581	2.215	232.48	4503	1.886	231.89	4436	1.638	231.30	4377
200	2.722	235.32	4616	2.254	234.77	4538	1.920	234.21	4471	1.668	233.65	4413
210	2.768	237.61	4650	2.293	237.08	4573	1.954	236.55	4506	1.698	236.01	4448
220	2.813	239.90	4684	2.331	239.40	4608	1.987	238.89	4541	1.728	238.37	4483
230	2.858	242.24	4718	2.369	241.75	4642	2.021	241.26	4576	1.758	240.77	4518
240	2.903	244.58	4752	2.407	244.11	4676	2.053	243.64	4610	1.788	243.16	4552
250	2.948	246.92	4785	2.445	246.47	4709	2.086	246.02	4644	1.817	245.56	4586
260	2.993	249.27	4818	2.483	248.83	4742	2.119	248.39	4677	1.846	247.95	4620
270	3.038	251.65	4851	2.521	251.22	4775	2.152	250.80	4710	1.875	250.37	4654
280	3.083	254.02	4884	2.559	253.61	4808	2.185	253.20	4743	1.904	252.79	4687
290	3.128	256.43	4916	2.597	256.03	4840	2.218	255.64	4776	1.933	255.24	4720
300	3.173	258.83	4948	2.634	258.46	4872	2.250	258.08	4808	1.961	257.69	4752
310	3.217	261.27	4980	2.671	260.91	4904	2.282	260.64	4840	1.989	260.17	4784
320	3.261	263.71	5011	2.708	263.35	4936	2.314	263.00	4872	2.017	262.64	4816
330	3.305	266.18	5043	2.745	265.83	4967	2.346	265.49	4904	2.045	265.14	4848
340	3.349	268.65	5074	2.782	268.31	4999	2.378	267.98	4935	2.073	267.64	4879
350	3.393	271.17	5105	2.819	270.82	5030	2.410	270.50	4966	2.101	270.17	4911
360	2.856	273.33	5061	2.442	273.02	4997	2.129	272.70	4942
370	2.474	275.57	5028	2.157	275.28	4973

TABLE A.5—METHYL CHLORIDE
PROPERTIES OF SUPERHEATED VAPOR (CONT.)

Temp. F.	Abs. Pressure, 100 lbs./in. ² Gage Pressure, 85.3 lbs./in. ² (Sat. Temp., 89.6 F.)				Abs. Pressure, 120 lbs./in. ² Gage Pressure, 105.3 lbs./in. ² (Sat. Temp., 101.9 F.)				Abs. Pressure, 140 lbs./in. ² Gage Pressure, 125.3 lbs./in. ² (Sat. Temp., 112.9 F.)				Abs. Pressure, 160 lbs./in. ² Gage Pressure, 145.3 lbs./in. ² (Sat. Temp., 122.8 F.)			
	t	v	h	s	t	v	h	s	t	v	h	s	t	v	h	s
Sat.		1.025	206.11	.3872		.858	207.10	.3827		.758	207.91	.3780		.647	208.59	.3757
90		1.026	206.21	.3877												
100		1.055	208.58	.3920												
110		1.083	210.96	.3962		.878	209.12	.3867								
120		1.111	213.33	.4003		.903	211.61	.3910								
130		1.138	215.74	.4044		.927	214.09	.3952		.754	209.70	.3825		.661	210.54	.3793
140		1.165	218.15	.4084		.951	216.57	.3993		.776	212.32	.3868		.681	213.24	.3837
150		1.191	220.55	.4124		.974	219.04	.4034		.818	217.48	.3954		.700	215.87	.3881
160		1.217	222.94	.4163		.996	221.51	.4074		.838	220.03	.3995		.719	218.50	.3924
170		1.243	225.33	.4201		1.019	223.97	.4113		.858	222.56	.4036		.737	221.11	.3966
180		1.268	227.71	.4239		1.041	226.42	.4152		.877	225.09	.4076		.754	223.73	.4007
190		1.293	230.10	.4276		1.062	228.88	.4190		.896	227.61	.4115		.772	226.31	.4048
200		1.318	232.50	.4312		1.083	231.33	.4228		.915	230.13	.4154		.789	228.90	.4087
210		1.343	234.91	.4349		1.104	233.79	.4265		.934	232.65	.4192		.806	231.48	.4126
220		1.367	237.32	.4384		1.125	236.25	.4301		.953	235.17	.4229		.823	234.06	.4164
230		1.391	239.76	.4420		1.146	238.74	.4337		.971	237.70	.4266		.839	236.63	.4202
240		1.415	242.20	.4455		1.167	241.22	.4373		.989	240.23	.4302		.855	239.21	.4239
250		1.439	244.63	.4489		1.187	243.69	.4408		1.007	242.73	.4338		.872	241.76	.4275
260		1.463	247.06	.4523		1.207	246.16	.4443		1.024	245.24	.4373		.888	244.30	.4311
270		1.487	249.51	.4557		1.227	248.64	.4477		1.042	247.76	.4407		.903	246.86	.4346
280		1.511	251.96	.4591		1.247	251.12	.4511		1.060	250.28	.4442		.919	249.42	.4381
290		1.534	254.44	.4624		1.267	253.64	.4545		1.077	252.82	.4476		.934	251.99	.4415
300		1.557	256.92	.4657		1.287	256.15	.4578		1.094	255.36	.4510		.950	254.56	.4449
310		1.580	259.43	.4689		1.307	258.67	.4611		1.111	257.91	.4543		.965	257.14	.4483
320		1.603	261.93	.4722		1.327	261.20	.4643		1.128	260.46	.4576		.980	259.72	.4516
330		1.626	264.45	.4754		1.346	263.75	.4676		1.145	263.04	.4609		.995	262.32	.4549
340		1.649	266.97	.4786		1.365	266.29	.4708		1.162	265.61	.4641		1.010	264.92	.4582
350		1.672	269.52	.4817		1.384	268.87	.4740		1.179	268.21	.4673		1.025	267.54	.4614
360		1.695	272.07	.4849		1.403	271.44	.4771		1.196	270.80	.4705		1.040	270.15	.4647
370		1.717	274.66	.4880		1.422	274.03	.4803		1.213	273.41	.4737		1.055	272.78	.4679
380		1.739	277.25	.4911		1.441	276.63	.4834		1.230	276.02	.4768		1.070	275.42	.4710
390		1.761	279.87	.4942		1.460	279.25	.4865		1.247	278.66	.4799		1.085	278.07	.4742
400			1.479	281.87	.4896		1.264	281.30	.4830		1.100	280.73	.4773
410			1.498	284.52	.4926		1.280	284.01	.4861		1.114	283.44	.4805
420			1.296	286.71	.4892		1.128	286.16	.4836

Temp. F.	Abs. Pressure, 180 lbs./in. ² Gage Pressure, 165.3 lbs./in. ² (Sat. Temp., 131.9 F.)			Abs. Pressure, 200 lbs./in. ² Gage Pressure, 185.3 lbs./in. ² (Sat. Temp., 140.3 F.)			Abs. Pressure, 220 lbs./in. ² Gage Pressure, 205.3 lbs./in. ² (Sat. Temp., 148.0 F.)			Abs. Pressure, 260 lbs./in. ² Gage Pressure, 245.3 lbs./in. ² (Sat. Temp., 162.3 F.)		
	.575	.599.15	.3788	.517	.599.60	.3708	.469	.599.98	.5678	.596	.510.62	.5637
130												
130												
140	.590	211.42	.3768									
150	.608	214.17	.3814	.533	212.41	.3749	.472	210.58	.3690			
160	.626	216.92	.3858	.551	215.20	.3796	.489	213.60	.3739			
170	.642	219.62	.3902	.566	218.09	.3842	.504	216.49	.3785	.406	213.04	.3679
180	.658	222.32	.3944	.582	220.87	.3886	.519	219.38	.3831	.421	216.18	.3728
190	.675	224.98	.3986	.597	223.61	.3929	.533	222.20	.3875	.434	219.20	.3775
200	.691	227.64	.4027	.612	226.35	.3971	.547	225.02	.3918	.447	222.22	.3820
210	.706	230.28	.4067	.626	229.05	.4012	.561	227.79	.3960	.459	225.15	.3865
220	.722	232.92	.4106	.641	231.75	.4052	.574	230.56	.4001	.471	228.08	.3908
230	.737	235.55	.4145	.654	234.43	.4091	.587	233.30	.4041	.483	230.95	.3950
240	.752	238.17	.4182	.668	237.12	.4129	.600	236.03	.4080	.495	233.81	.3991
250	.766	240.77	.4219	.682	239.75	.4167	.613	238.72	.4118	.506	236.60	.4031
260	.781	243.36	.4255	.695	242.39	.4204	.625	241.40	.4156	.517	239.38	.4070
270	.795	245.95	.4291	.708	245.02	.4240	.632	244.07	.4193	.528	242.14	.4108
280	.809	248.54	.4326	.721	247.65	.4276	.650	246.74	.4230	.539	244.90	.4145
290	.823	251.15	.4361	.734	250.29	.4311	.662	249.42	.4265	.549	247.66	.4182
300	.837	253.75	.4395	.747	252.93	.4346	.674	252.10	.4301	.560	250.41	.4218
310	.851	256.36	.4429	.760	255.58	.4380	.685	254.78	.4336	.570	253.15	.4254
320	.865	258.97	.4463	.772	258.21	.4414	.697	257.45	.4370	.581	255.89	.4289
330	.878	261.60	.4496	.785	260.86	.4448	.708	260.12	.4404	.591	258.62	.4324
340	.892	264.22	.4529	.797	263.51	.4481	.720	262.80	.4438	.601	261.35	.4359
350	.906	266.86	.4562	.810	266.17	.4515	.731	265.48	.4471	.611	264.09	.4393
360	.919	269.50	.4595	.822	268.84	.4547	.743	268.17	.4504	.621	266.82	.4426
370	.933	272.15	.4627	.834	271.51	.4580	.754	270.87	.4537	.631	269.56	.4460
380	.946	274.80	.4659	.847	274.19	.4612	.765	273.56	.4569	.640	272.31	.4493
390	.959	277.48	.4691	.859	276.88	.4644	.776	276.28	.4601	.650	275.05	.4525
400	.972	280.15	.4722	.870	279.59	.4676	.787	278.99	.4633	.659	277.80	.4558
410	.985	282.88	.4753	.883	282.31	.4707	.798	281.74	.4665	.669	280.60	.4590
420	.999	285.60	.4784	.895	285.05	.4738	.810	284.50	.4696	.679	283.39	.4621
430	1.011	288.33	.4815	.906	287.79	.4769	.820	287.25	.4727	.688	286.17	.4653
440	1.024	291.05	.4845	.918	290.52	.4800	.831	290.00	.4758	.697	288.96	.4684

TABLE A.6—DICHLORODIFLUOROMETHANE (FREON-12)
PROPERTIES OF LIQUID AND SATURATED VAPOR†

Temp.	Pressure		Volume		Density		Enthalpy from -40 F			Entropy from -40 F	
F	Abs. lb/in. ²	Gage lb/in. ²	Liquid ft ³ /lb	Vapor ft ³ /lb	Liquid lb/ft. ³	Vapor lb/ft. ³	Liquid Btu/lb	Latent Btu/lb	Vapor Btu/lb	Liquid Btu/lb F	Vapor Btu/lb F
t	p	p _g	v _f	v _g	1/v _f	1/v _g	h _f	h _{fg}	h _g	s _f	s _g
-155	0.1163	29.68*	0.00954	232.29	104.86	0.004305	-24.61	84.61	60.00	-0.0686	0.2092
-150	0.1527	29.61*	0.00957	179.79	104.46	0.005562	-23.50	84.07	60.57	-0.0650	0.2065
-145	.1985	29.52*	.00961	140.52	104.05	.007117	-22.39	83.53	61.14	-.0615	.2040
-140	.2554	29.40*	.00965	110.92	103.64	.009016	-21.29	83.01	61.72	-.0580	.2017
-135	.3256	29.28*	.00969	88.34	103.22	.01132	-20.19	82.49	62.30	-.0546	.1995
-130	.4116	29.08*	.00973	70.94	102.80	.01410	-19.10	81.98	62.88	-.0512	.1975
-125	0.5160	28.87*	0.00977	57.42	102.38	0.01742	-18.02	81.48	63.46	-0.0480	0.1955
-120	.6417	28.61*	.00981	46.84	101.95	.02135	-16.94	80.98	64.04	-.0448	.1937
-115	.7921	28.31*	.00985	38.49	101.52	.02598	-15.85	80.48	64.63	-.0416	.1919
-110	.9709	27.94*	.00989	31.84	101.08	.03141	-14.78	80.00	65.22	-.0385	.1903
-105	1.182	27.51*	.00994	26.51	100.64	.03773	-13.71	79.52	65.81	-.0355	.1888
-100	1.430	27.01*	0.00998	22.20	100.20	0.04504	-12.64	79.04	66.40	-0.0325	0.1873
-95	1.719	26.42*	.01003	18.71	99.75	.05344	-11.58	78.57	66.99	-.0295	.1860
-90	2.054	25.74*	.01007	15.86	99.30	.06305	-10.51	78.10	67.59	-.0266	.1847
-85	2.441	24.95*	.01012	13.51	98.85	.07400	-9.46	77.64	68.18	-.0238	.1835
-80	2.885	24.05*	.01016	11.57	98.39	.08640	-8.40	77.17	68.77	-.0210	.1823
-75	3.393	23.01*	0.01021	9.958	97.92	0.1004	-7.35	76.71	69.36	-0.0182	0.1813
-70	3.971	21.84*	.01026	8.608	97.46	.1162	-6.30	76.25	69.95	-.0155	.1802
-65	4.626	20.50*	.01031	7.474	96.99	.1338	-5.25	75.79	70.54	-.0128	.1793
-60	5.365	19.00*	.01036	6.516	96.51	.1535	-4.20	75.33	71.13	-.0102	.1783
-55	6.195	17.31*	.01041	5.704	96.04	.1753	-3.15	74.87	71.72	-.0076	.1774
-50	7.125	15.42*	0.01047	5.012	95.55	0.1995	-2.11	74.42	72.31	-0.0050	0.1767
-45	8.163	13.31*	.01052	4.420	95.07	.2263	-1.06	73.97	72.91	-.0025	.1759
-40	9.32	10.92*	0.0106	3.911	94.58	0.2557	0.0	73.50	73.50	0.0	0.17517
-38	9.82	9.91*	.0106	3.727	94.39	.2683	0.40	73.34	73.74	0.00094	.17490
-36	10.34	8.87*	.0106	3.553	94.20	.2815	0.81	73.17	73.98	.00188	.17463
-34	10.87	7.80*	.0106	3.389	93.99	.2951	1.21	73.01	74.22	.00282	.17438
-32	11.43	6.66*	.0107	3.234	93.79	.3092	1.62	72.84	74.46	.00376	.17412
-30	12.02	5.45*	0.0107	3.088	93.59	0.3238	2.03	72.67	74.70	0.00471	0.17387
-28	12.62	4.23*	.0107	2.950	93.39	.3390	2.44	72.50	74.94	.00565	.17364
-26	13.26	2.93*	.0107	2.827	93.18	.3546	2.85	72.33	75.18	.00659	.17340
-24	13.90	1.63*	.0108	2.698	92.98	.3706	3.25	72.16	75.41	.00753	.17317
-22	14.58	0.24*	.0108	2.583	92.78	.3871	3.66	71.98	75.64	.00846	.17296
-20	15.28	0.58	0.0108	2.474	92.58	0.4042	4.07	71.80	75.87	0.00940	0.17275
-18	16.01	1.31	.0108	2.370	92.38	.4219	4.48	71.63	76.11	.01033	.17253
-16	16.77	2.07	.0108	2.271	92.18	.4403	4.89	71.45	76.34	.01126	.17232
-14	17.55	2.85	.0109	2.177	91.97	.4593	5.30	71.27	76.57	.01218	.17212
-12	18.37	3.67	.0109	2.088	91.77	.4789	5.72	71.09	76.81	.01310	.17194
-10	19.20	4.50	0.0109	2.003	91.57	0.4993	6.14	70.91	77.05	0.01403	0.17175
-8	20.08	5.38	.0109	1.922	91.35	.5203	6.57	70.72	77.29	.01496	.17158
-6	20.98	6.28	.0110	1.845	91.14	.5420	6.99	70.53	77.52	.01589	.17140
-4	21.91	7.21	.0110	1.772	90.93	.5644	7.41	70.34	77.75	.01682	.17123
-2	22.87	8.17	.0110	1.703	90.72	.5872	7.83	70.15	77.98	.01775	.17107
0	23.87	9.17	0.0110	1.637	90.52	0.6109	8.25	69.96	78.21	0.01869	0.17091
2	24.89	10.19	.0110	1.574	90.31	.6352	8.67	69.77	78.44	.01961	.17075
4	25.96	11.26	.0111	1.514	90.11	.6606	9.10	69.57	78.67	.02052	.17060
6	26.61	11.81	.0111	1.455	90.00	.6755	9.52	69.47	78.79	.02097	.17053
8	27.05	12.35	.0111	1.457	89.88	.6864	9.53	69.37	78.90	.02143	.17045
8	28.18	13.48	.0111	1.403	89.68	.7129	9.96	69.17	79.13	.02235	.17030
10	29.35	14.65	0.0112	1.351	89.45	0.7402	10.39	68.97	79.36	0.02328	0.17015
12	30.56	15.86	.0112	1.301	89.24	.7687	10.82	68.77	79.59	.02419	.17001
14	31.80	17.10	.0112	1.253	89.03	.7981	11.26	68.56	79.82	.02510	.16987
16	33.08	18.38	.0112	1.207	88.81	.8288	11.70	68.35	80.05	.02601	.16974
18	34.40	19.70	.0113	1.163	88.58	.8598	12.12	68.15	80.27	.02692	.16961
20	35.75	21.05	0.0113	1.121	88.37	0.8921	12.55	67.94	80.49	0.02783	0.16949
22	37.15	22.45	.0113	1.081	88.13	.9251	13.00	67.72	80.72	.02873	.16938
24	38.58	23.88	.0113	1.043	87.91	.9588	13.44	67.51	80.95	.02963	.16926
26	40.07	25.37	.0114	1.007	87.68	.9930	13.88	67.29	81.17	.03053	.16913
28	41.59	26.89	.0114	0.973	87.47	1.028	14.32	67.07	81.39	.03143	.16900

* Inches of mercury below one atmosphere.

† Standard ton temperature.

‡ Courtesy Kinetic Chemicals, Inc.

TABLE A.6—DICHLORODIFLUOROMETHANE (FREON-12)
PROPERTIES OF LIQUID AND SATURATED VAPOR (CONT.)

Temp.	Pressure		Volume		Density		Enthalpy from -40 F			Entropy from -40 F	
F	Abs. lb/in ²	Gage lb/in ²	Liquid ft ³ /lb	Vapor ft ³ /lb	Liquid lb/ft ³	Vapor lb/ft ³	Liquid Btu/lb	Latent Btu/lb	Vapor Btu/lb	Liquid Btu/lb F	Vapor Btu/lb F
<i>t</i>	<i>p</i>	<i>p_g</i>	<i>v_f</i>	<i>v_g</i>	<i>1/v_f</i>	<i>1/v_g</i>	<i>h_f</i>	<i>h_{fg}</i>	<i>h_g</i>	<i>s_f</i>	<i>s_g</i>
30	43.16	28.46	0.0115	0.939	87.24	1.065	14.76	66.85	81.61	0.03233	0.16887
32	44.77	30.07	0.0115	.908	87.02	1.102	15.21	66.62	81.83	.03323	.16876
34	46.42	31.72	0.0115	.877	86.78	1.140	15.65	66.40	82.05	.03413	.16865
36	48.13	33.43	0.0116	.848	86.55	1.180	16.10	66.17	82.27	.03502	.16854
38	49.88	35.18	0.0116	.819	86.33	1.221	16.55	65.94	82.49	.03591	.16843
40	51.68	36.98	0.0116	0.792	86.10	1.263	17.00	65.71	82.71	0.03680	0.16833
42	53.51	38.81	0.0116	.767	85.88	1.304	17.46	65.47	82.93	.03770	.16823
44	55.40	40.70	0.0117	.742	85.66	1.349	17.91	65.24	83.15	.03859	.16813
46	57.35	42.65	0.0117	.718	85.43	1.393	18.36	65.00	83.36	.03948	.16803
48	59.35	44.65	0.0117	.695	85.19	1.438	18.82	64.74	83.57	.04037	.16794
50	61.39	46.69	0.0118	0.673	84.94	1.485	19.27	64.51	83.78	0.04126	0.16785
52	63.49	48.79	0.0118	.652	84.71	1.534	19.72	64.27	83.99	.04215	.16776
54	65.63	50.93	0.0118	.632	84.50	1.583	20.18	64.02	84.20	.04304	.16767
56	67.84	53.14	0.0119	.612	84.28	1.633	20.64	63.77	84.41	.04392	.16758
58	70.10	55.40	0.0119	.593	84.04	1.686	21.11	63.51	84.62	.04480	.16749
60	72.41	57.71	0.0119	0.575	83.78	1.740	21.57	63.25	84.82	0.04568	0.16741
62	74.77	60.07	0.0120	.557	83.57	1.795	22.03	62.99	85.02	.04657	.16733
64	77.20	62.50	0.0120	.540	83.34	1.851	22.49	62.73	85.22	.04745	.16725
66	79.67	64.97	0.0120	.524	83.10	1.909	22.95	62.47	85.42	.04833	.16717
68	82.24	67.54	0.0121	.508	82.86	1.968	23.42	62.20	85.62	.04921	.16709
70	84.82	70.12	0.0121	0.493	82.60	2.028	23.90	61.92	85.82	0.05009	0.16701
72	87.50	72.80	0.0121	.479	82.37	2.090	24.37	61.65	86.02	.05097	.16693
74	90.20	75.50	0.0122	.464	82.12	2.153	24.84	61.38	86.22	.05185	.16685
76	93.00	78.30	0.0122	.451	81.87	2.218	25.32	61.10	86.42	.05272	.16677
78	95.85	81.15	0.0123	.438	81.62	2.284	25.80	60.81	86.61	.05359	.16669
80	98.76	84.06	0.0123	0.425	81.39	2.353	26.28	60.52	86.80	0.05446	0.16662
82	101.7	87.00	0.0123	.413	81.12	2.423	26.76	60.23	86.99	.05534	.16655
84	104.8	90.1	0.0124	.401	80.87	2.495	27.24	59.94	87.18	.05621	.16648
86†	107.9	93.2	0.0124	.389	80.63	2.569	27.72	59.65	87.37	.05708	.16640
88	111.1	96.4	0.0124	.378	80.37	2.645	28.21	59.35	87.56	.05795	.16632
90	114.3	99.6	0.0125	0.368	80.11	2.721	28.70	59.04	87.74	0.05882	0.16624
92	117.7	103.0	0.0125	.357	79.86	2.799	29.19	58.73	87.92	.05969	.16616
94	121.0	106.3	0.0126	.347	79.60	2.880	29.68	58.42	88.10	.06056	.16608
96	124.5	109.8	0.0126	.338	79.32	2.963	30.18	58.10	88.28	.06143	.16600
98	128.0	113.3	0.0126	.328	79.06	3.048	30.67	57.78	88.45	.06230	.16592
100	131.6	116.9	0.0127	0.319	78.80	3.135	31.16	57.46	88.62	0.06316	0.16584
102	135.3	120.6	0.0127	.310	78.54	3.224	31.65	57.14	88.79	.06403	.16576
104	139.0	124.3	0.0128	.302	78.27	3.316	32.15	56.80	88.95	.06490	.16568
106	142.8	128.1	0.0128	.293	78.00	3.411	32.65	56.46	89.11	.06577	.16560
108	146.8	132.1	0.0129	.285	77.73	3.509	33.15	56.12	89.27	.06663	.16551
110	150.7	136.0	0.0129	0.277	77.46	3.610	33.65	55.78	89.43	0.06749	0.16542
112	154.8	140.1	0.0130	.269	77.18	3.714	34.15	55.43	89.58	.06836	.16533
114	158.9	144.2	0.0130	.262	76.89	3.823	34.65	55.08	89.73	.06922	.16524
116	163.1	148.4	0.0131	.254	76.60	3.934	35.15	54.72	89.87	.07008	.16515
118	167.4	152.7	0.0131	.247	76.32	4.049	35.65	54.36	90.01	.07094	.16506
120	171.8	157.1	0.0132	0.240	76.02	4.167	36.16	53.99	90.15	0.07180	0.16496
122	176.2	161.5	0.0132	.233	75.72	4.288	36.66	53.62	90.28	.07266	.16484
124	180.8	166.1	0.0133	.227	75.40	4.413	37.16	53.24	90.40	.07352	.16473
126	185.4	170.7	0.0133	.220	75.10	4.541	37.67	52.85	90.52	.07437	.16462
128	190.1	175.4	0.0134	.214	74.78	4.673	38.18	52.46	90.64	.07522	.16450
130	194.9	180.2	0.0134	0.208	74.46	4.808	38.69	52.07	90.76	0.07607	0.16438
132	199.8	185.1	0.0135	.202	74.13	4.948	39.19	51.67	90.86	.07691	.16425
134	204.8	190.1	0.0135	.196	73.81	5.094	39.70	51.26	90.96	.07775	.16411
136	209.9	195.2	0.0136	.191	73.46	5.247	40.21	50.85	91.06	.07858	.16396
138	215.0	200.3	0.0137	.185	73.10	5.405	40.72	50.43	91.15	.07941	.16380
140	220.2	205.5	0.0138	0.180	72.73	5.571	41.24	50.00	91.24	0.08024	0.16363

† Standard ton temperature.

TABLE A.7—DICHLORODIFLUOROMETHANE (FREON-12)
PROPERTIES OF SUPERHEATED VAPOR*

Temp. F.	Abs. Pressure 0.14 lbs./in. ² Gage Pressure 29.64 in. vac. (Sat. Temp. -151.7 F.)				Abs. Pressure 0.20 lbs./in. ² Gage Pressure 29.51 in. vac. (Sat. Temp. -144.9 F.)				Abs. Pressure 0.40 lbs./in. ² Gage Pressure 29.11 in. vac. (Sat. Temp. -130.7 F.)				Abs. Pressure 0.60 lbs./in. ² Gage Pressure 28.70 in. vac. (Sat. Temp. -121.7 F.)			
	<i>t</i>	<i>p</i>	<i>h</i>	<i>s</i>	<i>p</i>	<i>h</i>	<i>s</i>	<i>p</i>	<i>h</i>	<i>s</i>	<i>p</i>	<i>h</i>	<i>s</i>	<i>p</i>	<i>h</i>	<i>s</i>
Sat.		195.0	60.57	0.2075	159.5	61.16	0.2040	72.84	62.80	0.1978	49.85	63.85	0.1945			
-150		196.1	60.57	0.2080	141.7	61.72	0.2059	73.00	62.83	0.1980						
-140		202.5	61.73	0.2117	146.1	62.89	0.2094	75.22	64.06	0.2015	50.11	64.04	0.1948			
-130		208.8	62.89	0.2153	150.6	64.07	0.2129	77.45	65.25	0.2060	51.59	65.24	0.1983			
-120		215.2	64.08	0.2188	155.0	65.27	0.2164									
-110		221.5	65.27	0.2222	159.5	66.47	0.2198	79.67	66.46	0.2084	53.08	66.45	0.2017			
-100		227.9	66.48	0.2256	163.9	67.70	0.2231	81.90	67.68	0.2117	54.56	67.67	0.2050			
-90		234.2	67.70	0.2290	168.3	68.93	0.2264	84.12	68.92	0.2150	56.05	68.91	0.2083			
-80		240.5	68.93	0.2323	172.8	70.18	0.2297	86.34	70.17	0.2183	57.53	70.16	0.2116			
-70		246.9	70.18	0.2355	177.2	71.44	0.2329	88.57	71.43	0.2215	59.01	71.42	0.2148			
-60		253.2	71.44	0.2387	181.7	72.71	0.2360	90.79	72.70	0.2246	60.50	72.69	0.2179			
-50		259.6	72.72	0.2419	186.1	74.00	0.2391	93.01	73.99	0.2277	61.98	73.98	0.2210			
-40		265.9	74.00	0.2450	190.6	75.30	0.2422	95.24	75.29	0.2308	63.46	75.28	0.2241			
-30		272.3	75.31	0.2480	195.0	76.62	0.2452	97.46	76.61	0.2338	64.95	76.60	0.2271			
-20		278.6	76.62	0.2511	199.4	77.94	0.2482	99.68	77.93	0.2368	66.43	77.93	0.2301			
-10		284.9	77.95	0.2541	203.9	79.29	0.2511	101.9	79.28	0.2397	67.91	79.27	0.2331			
0		291.3	79.29	0.2570	208.3	80.64	0.2541	104.1	80.63	0.2427	69.39	80.62	0.2360			
10		297.6	80.64	0.2599	212.8	82.01	0.2569	106.4	82.00	0.2455	70.88	81.99	0.2389			
20		304.0	82.01	0.2628	217.2	83.39	0.2598	108.6	83.38	0.2484	72.36	83.37	0.2417			
30		310.3	83.39	0.2656	221.6	84.78	0.2626	110.8	84.77	0.2512	73.84	84.77	0.2445			
40		316.7	84.78	0.2685	226.1	86.19	0.2654	113.0	86.18	0.2540	75.33	86.17	0.2473			
50		323.0	86.19	0.2713	230.5	87.61	0.2682	115.2	87.60	0.2568	76.81	87.59	0.2501			
60		329.3	87.61	0.2740	235.0	89.04	0.2709	117.5	89.04	0.2595	78.29	89.03	0.2528			
70		335.7	89.05	0.2767	239.4	90.49	0.2736	119.7	90.48	0.2622	79.77	90.48	0.2555			
80		342.0	90.49	0.2795	243.8	91.95	0.2763	121.9	91.94	0.2649	81.25	91.94	0.2582			
90		348.4	91.95	0.2821	248.3	93.42	0.2789	124.1	93.42	0.2675	82.73	93.41	0.2609			
100		354.7	93.43	0.2848	252.7	94.91	0.2816	126.3	94.91	0.2702	84.22	94.90	0.2635			
110		361.1	94.91	0.2874	257.2	96.41	0.2842	128.6	96.41	0.2728	85.70	96.40	0.2661			
120		367.4	96.41	0.2901	261.6	97.93	0.2868	130.8	97.92	0.2754	87.18	97.91	0.2687			
130		373.7	97.93	0.2926	266.0	99.45	0.2893	133.0	99.45	0.2779	88.66	99.44	0.2713			
140		380.1	99.45	0.2952	270.5	100.99	0.2919	135.2	100.99	0.2805	90.14	100.98	0.2738			
150		386.4	101.00	0.2978	274.9	102.55	0.2944	137.5	102.54	0.2830	91.62	102.54	0.2764			
160								139.7	104.11	0.2855	93.10	104.10	0.2789			
170											94.59	105.68	0.2814			
180																

Temp. F.	Abs. Pressure 0.80 lbs./in. ² Gage Pressure 28.29 in. vac. (Sat. Temp. -114.8 F.)		Abs. Pressure 1.00 lbs./in. ² Gage Pressure 27.88 in. vac. (Sat. Temp. -109.3 F.)		Abs. Pressure 2.0 lbs./in. ² Gage Pressure 26.85 in. vac. (Sat. Temp. -90.8 F.)		Abs. Pressure 3.0 lbs./in. ² Gage Pressure 23.81 in. vac. (Sat. Temp. -78.9 F.)	
Sat.	58.13	64.66	65.90	66.49	66.49	66.49	68.90	11.16
-110	38.66	65.23						
-100	39.78	66.44	66.43	67.59	67.59	67.59		
-90	40.89	67.66	67.65	68.83	68.83	68.83		
-80	42.01	68.90	68.89	70.13	70.08	70.08	70.00	0.1849
-70	43.12	70.14	70.13	71.40	71.35	71.35	71.27	0.1881
-60	44.24	71.41	71.40	72.67	72.62	72.62		
-50	45.35	72.68	72.67	73.96	73.91	73.91	72.55	0.1913
-40	46.47	73.97	73.96	75.26	75.21	75.21	73.84	0.1944
-30	47.58	75.27	75.26	76.58	76.53	76.53	75.15	0.1975
-20	48.69	76.59	76.58	77.91	77.86	77.86	76.47	0.2005
-10	49.81	77.92	77.91	79.25	79.21	79.21	77.80	0.2035
0	50.92	79.26	79.25	80.61	80.56	80.56	79.15	0.2065
10	52.03	80.61	80.61	81.97	81.93	81.93	80.51	0.2094
20	53.14	81.98	81.97	83.35	83.31	83.31	81.88	0.2123
30	54.26	83.36	83.35	84.75	84.71	84.71	83.26	0.2151
40	55.37	84.76	84.75	86.16	86.12	86.12	84.66	0.2180
50	56.48	86.17	86.16	87.58	87.54	87.54	86.07	0.2208
60	57.59	87.59	87.58	89.01	88.98	88.98	87.50	0.2235
70	58.71	89.02	89.01	90.46	90.43	90.43	88.93	0.2263
80	59.82	90.47	90.46	91.92	91.89	91.89	90.38	0.2290
90	60.94	91.93	91.92	93.40	93.37	93.37	91.85	0.2317
100	62.04	93.40	93.40	94.89	94.85	94.85	93.32	0.2343
110	63.15	94.89	94.89	96.39	96.36	96.36	94.81	0.2370
120	64.27	96.39	96.39	97.90	97.87	97.87	96.32	0.2396
130	65.38	97.91	97.90	99.43	99.40	99.40	97.83	0.2422
140	66.49	99.44	99.43	100.97	100.94	100.94	99.36	0.2448
150	67.60	100.98	100.97	102.52	102.50	102.50	100.90	0.2473
160	68.71	102.53	102.52	104.09	104.06	104.06	102.46	0.2498
170	69.82	104.10	104.09	105.67	105.65	105.65	104.04	0.2523
180	70.93	105.68	105.67	107.27	107.24	107.24	105.62	0.2548
190	72.04	107.27	107.27	108.87	108.85	108.85	107.21	0.2573
200			58.52	108.87	108.85	108.85	108.82	0.2598
210					110.47	110.47	110.45	0.2622
220							112.08	0.2646
230							113.73	0.2670

* Courtesy Kinetic Chemicals, Inc.

TABLE A.7—DICHLORODIFLUOROMETHANE (FREON-12)
PROPERTIES OF SUPERHEATED VAPOR (CONT.)

Temp. F.	Abs. Pressure 5.0 lb./in. ² Gage Pressure 19.74 in. vac. (Sat. Temp. -62.5 F.)				Abs. Pressure 7.5 lb./in. ² Gage Pressure 14.65 in. vac. (Sat. Temp. -48.6 F.)				Abs. Pressure 10.0 lb./in. ² Gage Pressure 9.57 in. vac. (Sat. Temp. -37.3 F.)				Abs. Pressure 15 lb./in. ² Gage Pressure 0.3 lb./in. ² (Sat. Temp. -20.8 F.)			
	<i>t</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>t</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>t</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>t</i>	<i>v</i>	<i>h</i>	<i>s</i>
Sat.		6.953	70.83	0.1788		4.774	72.47	0.1765		3.662	73.82	0.1748		2.518	75.78	0.1728
-60		6.908	71.15	0.1796												
-50		7.180	72.43	.1827		4.879	73.60	0.1791		3.728	74.77	0.1770		2.521	75.89	0.17307
-40		7.363	73.72	.1859		5.002	74.90	.1820		3.821	76.11	.1801		2.583	77.23	.17611
-30		7.544	75.03	.1889		5.125	76.23	.1850		3.913	77.46	.1831				
-20		7.726	76.36	.1920		5.247	77.56	.1880								
-10		7.907	77.70	.1950												
0		8.088	79.05	0.1979		5.368	78.92	0.1910		4.006	78.81	0.1861		2.646	78.59	0.17913
10		8.269	80.41	.2009		5.490	80.28	.1940		4.098	80.18	.1891		2.708	79.97	.18208
20		8.450	81.78	.2038		5.612	81.67	.1969		4.189	81.56	.1919		2.771	81.37	.18499
30		8.631	83.16	.2066		5.733	83.06	.1997		4.280	82.94	.1948		2.833	82.77	.18788
40		8.812	84.56	.2095		5.854	84.46	.2026		4.371	84.35	.1977		2.895	84.18	.19074
50		8.992	85.98	0.2123		5.976	85.88	0.2054		4.463	85.77	0.2005		2.957	85.60	0.19357
60		9.173	87.41	.2150		6.097	87.30	.2081		4.556	87.19	.2033		3.019	87.03	.19635
70		9.353	88.85	.2178		6.218	88.75	.2109		4.648	88.64	.2060		3.081	88.48	.19911
80		9.533	90.30	.2205		6.339	90.20	.2136		4.740	90.11	.2087		3.143	89.94	.20185
90		9.713	91.77	.2232		6.459	91.68	.2163		4.832	91.58	.2114		3.204	91.41	.20455
100		9.893	93.25	0.2258		6.581	93.15	0.2190		4.923	93.05	0.2141		3.266	92.91	0.20723
110		10.07	94.74	.2285		6.702	94.65	.2216		5.015	94.56	.2168		3.327	94.41	.20989
120		10.25	96.24	.2311		6.822	96.15	.2243		5.107	96.07	.2194		3.388	95.91	.21252
130		10.43	97.76	.2337		6.942	97.68	.2269		5.198	97.59	.2220		3.450	97.44	.21513
140		10.61	99.29	.2363		7.062	99.20	.2294		5.289	99.14	.2246		3.510	98.98	.21772
150		10.79	100.84	0.2388		7.182	100.76	0.2320		5.379	100.66	0.2271		3.571	100.53	0.22028
160		10.97	102.39	.2414		7.303	102.31	.2345		5.470	102.24	.2297		3.632	102.10	.22282
170		11.15	103.96	.2439		7.423	103.89	.2370		5.560	103.81	.2322		3.694	103.68	.22535
180		11.33	105.55	.2464		7.543	105.47	.2395		5.650	105.40	.2347		3.755	105.27	.22786
190		11.51	107.14	.2489		7.663	107.07	.2420		5.740	107.00	.2372		3.816	106.87	.23034
200		11.69	108.75	0.2513		7.783	108.68	0.2445		5.831	108.63	0.2396		3.877	108.49	0.23282
210		11.87	110.38	.2538		7.904	110.31	.2469		5.921	110.25	.2421		3.938	110.12	.23527
220		12.05	112.01	.2562		8.024	111.94	.2493		6.011	111.88	.2445		3.998	111.77	.23770
230		12.23	113.66	.2586		8.144	113.60	.2518		6.101	113.53	.2469		4.059	113.42	.24011
240		12.41	115.33	.2610		8.264	115.26	.2542		6.191	115.20	.2493		4.120	115.10	.24253
250						8.385	116.94	0.2565		6.281	116.88	0.2517		4.181	116.78	0.24491
260						8.504	118.63	.2589		6.371	118.58	.2541				
270										6.461	120.30	.2564				

APPENDIX

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Temp. F.	Abs. Pressure 20 lb./in. ² Gage Pressure 5.3 lb./in. ² (Sat. Temp. -8.2 F.)			Abs. Pressure 26 lb./in. ² Gage Pressure 11.3 lb./in. ² (Sat. Temp. 4.1 F.)			Abs. Pressure 32 lb./in. ² Gage Pressure 17.3 lb./in. ² (Sat. Temp. 14.3 F.)			Abs. Pressure 40 lb./in. ² Gage Pressure 25.3 lb./in. ² (Sat. Temp. 25.9 F.)		
	1.925	77.27	0.17160	1.510	78.68	0.17059	1.245	79.84	0.16985	1.009	81.16	0.16914
0	1.965	78.39	0.17407	1.530	79.50	0.17233	1.262	80.67	0.17152	1.019	81.76	0.17030
10	2.013	79.76	.17704	1.568	80.90	.17529	1.293	82.08	.17445	1.044	83.20	.17322
20	2.060	81.14	.17996	1.606	82.31	.17821	1.323	83.51	.17734			
30	2.107	82.55	.18286	1.643	83.74	.18109						
40	2.155	83.97	.18573									
50	2.203	85.40	0.18858	1.680	85.19	0.18395	1.354	84.96	0.18022	1.070	84.65	0.17612
60	2.250	86.85	.19138	1.716	86.64	.18675	1.384	86.41	.18304	1.095	86.11	.17896
70	2.297	88.31	.19415	1.754	88.10	.18954	1.414	87.88	.18583	1.120	87.60	.18178
80	2.343	89.78	.19688	1.790	89.57	.19229	1.444	89.36	.18960	1.144	89.09	.18455
90	2.390	91.26	.19959	1.826	91.05	.19501	1.474	90.85	.19133	1.169	90.58	.18731
100	2.437	92.75	0.20229	1.863	92.55	0.19772	1.504	92.35	0.19404	1.194	92.09	0.19004
110	2.483	94.26	.20494	1.900	94.06	.20039	1.535	93.87	.19673	1.218	93.62	.19272
120	2.530	95.78	.20759	1.936	95.58	.20305	1.565	95.40	.19940	1.242	95.15	.19538
130	2.577	97.31	.21020	1.972	97.12	.20565	1.595	96.94	.20202	1.267	96.70	.19803
140	2.623	98.85	.21280	2.008	98.67	.20827	1.624	98.50	.20463	1.291	98.26	.20066
150	2.669	100.40	0.21537	2.045	100.23	0.21085	1.654	100.06	0.20721	1.315	99.83	0.20325
160	2.716	101.97	.21792	2.082	101.80	.21341	1.683	101.64	.20977	1.340	101.42	.20583
170	2.762	103.56	.22045	2.117	103.39	.21594	1.713	103.23	.21232	1.364	103.02	.20838
180	2.808	105.15	.22297	2.153	104.99	.21847	1.743	104.83	.21486	1.388	104.63	.21092
190	2.854	106.76	.22545	2.189	106.60	.22096	1.772	106.45	.21735	1.412	106.25	.21343
200	2.901	108.38	0.22794	2.224	108.23	0.22344	1.802	108.08	0.21985	1.435	107.88	0.21592
210	2.947	110.01	.23039	2.259	109.86	.22589	1.831	109.72	.22231	1.459	109.52	.21840
220	2.992	111.65	.23283	2.295	111.51	.22834	1.860	111.36	.22476	1.482	111.17	.22085
230	3.038	113.31	.23524	2.331	113.17	.23076	1.889	113.03	.22718	1.506	112.84	.22329
240	3.084	114.98	.23716	2.366	114.85	.23318	1.918	114.72	.22960	1.530	114.52	.22572
250	3.130	116.67	0.24005	2.402	116.54	0.23557	1.948	116.41	0.23200	1.554	116.21	0.22813
260	3.177	118.36	.24242	2.438	118.24	.23796	1.977	118.11	.23439	1.577	117.92	.23052
270	3.223	120.07	.24477	2.474	119.95	.24033	2.006	119.82	.23676	1.601	119.65	.23289
280	2.510	121.68	.24267	2.035	121.55	.23912	1.625	121.40	.23526
290	2.065	123.30	.24146	1.649	123.15	.23760

TABLE A.7—DICHLORODIFLUOROMETHANE (FREON-12)
PROPERTIES OF SUPERHEATED VAPOR (CONT.)

Temp F	Abs. Pressure 50 lb/in. ² Gage Pressure 35.3 lb/in. ² (Sat. Temp. 38.3 F.)				Abs. Pressure 60 lb/in. ² Gage Pressure 45.3 lb/in. ² (Sat. Temp. 48.7 F.)				Abs. Pressure 80 lb/in. ² Gage Pressure 65.3 lb/in. ² (Sat. Temp. 66.3 F.)				Abs. Pressure 100 lb/in. ² Gage Pressure 85.3 lb/in. ² (Sat. Temp. 80.9 F.)			
	t	h	s	r	t	h	s	r	t	h	s	r	t	h	s	r
Sat.	0.817	82.52	0.16841	0.688	83.65	0.16791	0.521	85.45	0.16716	0.419	86.89	0.16659				
40	0.821	82.76	0.16895													
50	0.842	84.24	0.17187	0.690	83.83	0.16829										
60	0.863	85.72	0.17475	0.708	85.33	0.17120										
70	0.884	87.22	0.17760	0.726	86.84	0.17407	0.526	86.01	0.16819							
80	0.904	88.72	0.18040	0.743	88.35	0.17689	0.540	87.56	0.17108							
90	0.924	90.23	0.18317	0.760	89.87	0.17968	0.554	89.12	0.17394							
100	0.944	91.75	0.18591	0.778	91.41	0.18246	0.568	90.68	0.17675							
110	0.964	93.29	0.18862	0.795	92.96	0.18519	0.582	92.26	0.17954							
120	0.984	94.83	0.19132	0.812	94.51	0.18789	0.596	93.84	0.18229							
130	1.004	96.39	0.19397	0.829	96.07	0.19056	0.609	95.43	0.18500							
140	1.024	97.96	0.19662	0.846	97.66	0.19323	0.623	97.03	0.18771							
150	1.044	99.54	0.19923	0.863	99.24	0.19585	0.636	98.64	0.19035							
160	1.064	101.14	0.20182	0.880	100.84	0.19846	0.649	100.26	0.19298							
170	1.084	102.75	0.20439	0.897	102.45	0.20104	0.662	101.88	0.19558							
180	1.103	104.36	0.20694	0.913	104.07	0.20360	0.675	103.52	0.19817							
190	1.123	105.98	0.20946	0.930	105.71	0.20613	0.688	105.18	0.20073							
200	1.142	107.62	0.21196	0.946	107.36	0.20865	0.701	106.84	0.20328							
210	1.162	109.28	0.21444	0.962	109.02	0.21113	0.714	108.51	0.20580							
220	1.181	110.95	0.21691	0.979	110.69	0.21361	0.726	110.19	0.20828							
230	1.200	112.62	0.21935	0.995	112.37	0.21607	0.739	111.88	0.21076							
240	1.220	114.31	0.22179	1.012	114.06	0.21853	0.751	113.58	0.21321							
250	1.239	116.00	0.22419	1.028	115.77	0.22094	0.764	115.30	0.21566							
260	1.258	117.71	0.22660	1.044	117.49	0.22334	0.777	117.03	0.21809							
270	1.277	119.44	0.22898	1.060	119.23	0.22573	0.789	118.78	0.22049							
280	1.296	121.18	0.23134	1.076	120.97	0.22810	0.802	120.54	0.22289							
290	1.314	122.93	0.23367	1.092	122.73	0.23045	0.814	122.30	0.22525							
300	1.332	124.69	0.23600	1.108	124.50	0.23280	0.826	124.08	0.22760							
310	1.350	126.45	0.23831	1.124	126.28	0.23513	0.839	125.88	0.22995							
320	1.140	128.07	0.23745	0.851	127.70	0.23229							
330	0.864	129.52	0.23461							
340							

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Temp. F.	Abs. Press. 120 lb/in. ² Gage Press. 105.3 lb/in. ² (Sat. Temp. 93.4 F.)			Abs. Press. 140 lb/in. ² Gage Press. 125.3 lb/in. ² (Sat. Temp. 104.5 F.)			Abs. Press. 180 lb/in. ² Gage Press. 165.3 lb/in. ² (Sat. Temp. 123.7 F.)			Abs. Press. 220 lb/in. ² Gage Press. 205.3 lb/in. ² (Sat. Temp. 139.9 F.)		
	0.550	88.05	0.16610	0.498	88.99	0.16566	0.228	90.38	0.16476	0.181	91.50	0.16375
90												
100	0.357	89.13	0.16803	0.304	89.92	0.16725	0.233	91.47	0.16665	0.181	91.52	0.16378
110	0.367	90.75	0.17090	0.314	91.60	0.17021	0.241	93.23	0.16964			
120	0.377	92.38	0.17374	0.323	93.28	0.17306						
130	0.387	94.01	0.17654	0.332	94.96	0.17590						
140	0.397	95.65	0.17932									
150	0.407	97.30	0.18207	0.341	96.65	0.17868	0.249	94.99	0.17254	0.188	93.32	0.16685
160	0.417	98.96	0.18474	0.350	98.34	0.18142	0.257	96.75	0.17541	0.195	95.13	0.16986
170	0.426	100.63	0.18743	0.358	100.03	0.18412	0.265	98.52	0.17823	0.202	96.94	0.17282
180	0.436	102.31	0.19011	0.366	101.72	0.18678	0.272	100.29	0.18102	0.209	98.75	0.17576
190	0.445	104.00	0.19271	0.374	103.42	0.18941	0.280	102.07	0.18377	0.216	100.57	0.17861
200	0.454	105.70	0.19529	0.383	105.14	0.19205	0.287	103.85	0.18648	0.223	102.39	0.18142
210	0.463	107.41	0.19785	0.391	106.86	0.19466	0.294	105.63	0.18912	0.229	104.22	0.18420
220	0.472	109.13	0.20041	0.399	108.59	0.19724	0.301	107.42	0.19174	0.236	106.05	0.18694
230	0.482	110.86	0.20294	0.407	110.33	0.19976	0.307	109.21	0.19433	0.242	107.89	0.18962
240	0.491	112.60	0.20545	0.415	112.09	0.20229	0.314	111.01	0.19693	0.248	109.74	0.19226
250	0.500	114.35	0.20792	0.423	113.85	0.20479	0.321	112.81	0.19947	0.254	111.59	0.19487
260	0.508	116.11	0.21035	0.431	115.63	0.20728	0.327	114.62	0.20199	0.259	113.44	0.19745
270	0.517	117.88	0.21279	0.439	117.42	0.20974	0.334	116.44	0.20449	0.265	115.30	0.20001
280	0.526	119.66	0.21521	0.447	119.22	0.21219	0.340	118.26	0.20698	0.271	117.16	0.20255
290	0.534	121.45	0.21760	0.455	121.03	0.21461	0.347	120.09	0.20944	0.277	119.03	0.20504
300	0.543	123.25	0.22000	0.462	122.85	0.21701	0.353	121.92	0.21187	0.282	120.91	0.20753
310	0.552	125.07	0.22238	0.470	124.67	0.21939	0.359	123.76	0.21428	0.288	122.80	0.21002
320	0.560	126.90	0.22472	0.477	126.50	0.22174	0.365	125.61	0.21665	0.293	124.70	0.21244
330	0.569	128.74	0.22707	0.485	128.33	0.22411	0.371	127.47	0.21904	0.299	126.60	0.21485
340	0.578	130.59	0.22940	0.492	130.17	0.22646	0.377	129.34	0.22140	0.304	128.51	0.21724
350	0.586	132.45	0.23171	0.500	132.02	0.22880	0.383	131.23	0.22374	0.309	130.42	0.21960
360	0.595	134.32	0.23400	0.507	133.89	0.23109	0.390	133.13	0.22608	0.314	132.34	0.22195
370	0.515	135.78	0.23336	0.396	135.05	0.22840	0.320	134.27	0.22430
380	0.402	136.98	0.23072	0.325	136.21	0.22665
390	0.408	138.91	0.23301	0.330	138.16	0.22895
400	0.414	140.85	0.23529	0.335	140.12	0.23124

TABLE A.8—ETHYLENE
PROPERTIES OF LIQUID AND SATURATED VAPOR†

Temp. F t	Pressure		Volume		Density		Enthalpy from -40 F			Entropy from -40 F	
	Abs. lb/in ² p	Gage lb/in ² p _g	Liquid ft ³ /lb v _f	Vapor ft ³ /lb v _g	Liquid lb/ft ³ 1/v _f	Vapor lb/ft ³ 1/v _g	Liquid Btu/lb h _f	Vapor Btu/lb h _g	Latent Btu/lb h _{fg}	Liquid Btu/lb F s _f	Vapor Btu/lb F s _g
-176.8	6.75	*16.18	0.02732	16.4	36.60	0.061	0	213.9	213.9	0.000	0.756
-173.2	7.72	*14.21	0.02746	14.5	36.42	0.069	1.8	214.5	212.7	0.006	0.749
-169.6	8.80	*12.01	0.02760	12.7	36.23	0.079	3.5	215.1	211.6	0.012	0.742
-166.0	10.00	*9.57	0.02774	11.1	36.05	0.090	5.3	215.8	210.5	0.018	0.735
-162.4	11.36	*7.02	0.02788	9.71	35.87	0.103	7.1	216.3	209.2	0.024	0.728
-158.8	12.80	*3.87	0.02802	8.62	35.69	0.116	9.0	216.9	207.9	0.030	0.722
-155.2	14.44	*0.52	0.02815	7.52	35.51	0.131	10.8	217.5	206.7	0.036	0.716
-151.6	16.22	1.52	0.02830	6.76	35.32	0.148	12.6	218.1	205.5	0.042	0.710
-148.0	18.17	3.47	0.02846	6.06	35.14	0.165	14.5	218.7	204.2	0.048	0.704
-144.4	20.36	5.66	0.02860	5.49	34.96	0.182	16.4	219.2	202.8	0.054	0.698
-140.8	22.65	7.95	0.02875	4.98	34.78	0.201	18.3	219.9	201.6	0.060	0.693
-137.2	25.17	10.47	0.02890	4.50	34.60	0.222	20.2	220.6	200.4	0.066	0.688
-133.6	27.91	13.21	0.02905	4.08	34.42	0.245	22.2	221.2	199.0	0.072	0.683
-130.0	30.86	16.16	0.02921	3.72	34.23	0.269	24.1	221.8	197.7	0.078	0.678
-126.4	34.06	19.36	0.02937	3.39	34.05	0.295	26.0	222.3	196.3	0.084	0.673
-122.8	37.50	22.80	0.02953	3.11	33.87	0.322	28.1	223.1	195.0	0.090	0.669
-119.2	41.19	26.49	0.02968	2.85	33.69	0.351	30.1	223.8	193.7	0.096	0.665
-115.6	45.16	30.46	0.02985	2.62	33.50	0.382	32.2	224.6	192.4	0.102	0.661
-112.0	49.40	34.70	0.03003	2.40	33.30	0.416	34.2	225.3	191.1	0.108	0.658
-108.4	53.93	39.23	0.03020	2.21	33.11	0.452	36.3	226.0	189.7	0.114	0.654
-104.8	58.76	44.06	0.03039	2.04	32.91	0.490	38.4	226.7	188.3	0.120	0.650
-101.2	63.90	49.20	0.03056	1.89	32.72	0.530	40.5	227.3	186.8	0.125	0.647
-97.6	69.37	54.67	0.03077	1.75	32.50	0.573	42.6	228.0	185.4	0.131	0.643
-94.0	75.17	60.47	0.03098	1.62	32.28	0.617	44.8	228.7	183.9	0.137	0.640
-90.4	81.32	66.62	0.03118	1.51	32.07	0.663	47.0	229.4	182.4	0.143	0.637
-86.8	87.88	73.18	0.03139	1.40	31.86	0.713	49.2	230.1	180.9	0.149	0.634
-83.2	94.76	80.06	0.03161	1.31	31.64	0.765	51.4	230.8	179.4	0.155	0.631
-79.6	102.0	87.3	0.03183	1.22	31.42	0.820	53.7	231.5	177.8	0.161	0.628
-76.0	109.7	95.0	0.03205	1.14	31.20	0.877	55.9	232.1	176.2	0.166	0.626
-72.4	117.7	103.0	0.03228	1.07	30.98	0.938	58.3	232.9	174.6	0.172	0.623
-68.8	126.2	111.5	0.03251	1.00	30.76	0.999	60.6	233.5	172.9	0.178	0.621
-65.2	135.1	120.4	0.03274	0.935	30.54	1.065	63.0	234.3	171.3	0.184	0.618
-61.6	144.4	129.7	0.03299	0.885	30.31	1.138	65.3	234.9	169.5	0.190	0.616
-58.0	154.2	139.5	0.03326	0.826	30.07	1.212	67.8	235.5	167.7	0.196	0.613
-54.4	164.5	149.8	0.03352	0.775	29.83	1.292	70.3	235.8	165.5	0.202	0.610
-50.8	175.2	160.5	0.03381	0.730	29.58	1.374	72.7	236.1	163.4	0.208	0.608
-47.2	186.7	172.0	0.03408	0.685	29.34	1.460	75.3	236.4	161.2	0.214	0.605
-43.6	198.1	183.4	0.03436	0.645	29.10	1.552	77.8	236.7	158.9	0.220	0.602
-40.0	210.4	196.7	0.03467	0.606	28.84	1.649	80.4	237.0	156.6	0.226	0.599
-36.4	223.2	208.5	0.03499	0.571	28.58	1.754	83.0	237.3	154.3	0.232	0.596
-32.8	236.6	221.9	0.03531	0.535	28.32	1.865	85.7	237.5	151.8	0.238	0.594
-29.2	250.5	235.8	0.03566	0.503	28.04	1.985	88.4	237.4	149.0	0.244	0.590
-25.6	265.2	250.5	0.03602	0.474	27.76	2.11	91.1	237.4	146.3	0.250	0.587
-22.0	280.1	265.4	0.03640	0.444	27.47	2.25	93.9	237.4	143.5	0.256	0.584
-18.4	296.0	281.3	0.03679	0.418	27.18	2.39	96.8	237.2	140.4	0.263	0.581
-14.8	312.2	297.5	0.03721	0.392	26.87	2.55	99.6	236.8	137.1	0.269	0.577
-11.2	329.3	314.6	0.03760	0.368	26.56	2.72	102.7	236.4	133.7	0.275	0.573
-7.6	346.8	332.1	0.03811	0.345	26.24	2.90	105.7	235.9	130.2	0.282	0.570
-4.0	365.5	350.8	0.03858	0.323	25.92	3.10	108.8	235.3	126.5	0.288	0.566
0.4	384.3	369.6	0.03911	0.302	25.57	3.31	112.0	234.8	122.8	0.295	0.562
+ 3.2	404.4	389.7	0.03965	0.283	25.22	3.53	115.2	234.2	118.9	0.301	0.558
6.8	424.7	410.0	0.04026	0.266	24.84	3.76	118.7	233.6	114.9	0.308	0.555
10.4	446.5	431.8	0.04090	0.249	24.45	4.01	122.3	233.0	110.6	0.316	0.551
14.0	468.7	454.0	0.04161	0.234	24.03	4.28	126.0	232.4	106.5	0.323	0.548
17.6	491.8	477.1	0.04237	0.218	23.60	4.58	130	232	101.9	0.331	0.545
21.2	515.0	500.3	0.04323	0.205	23.13	4.88	134	230	97.0	0.339	0.541
24.8	540.4	525.7	0.04419	0.190	22.63	5.26	138	229	91.9	0.348	0.537
28.4	566.7	551.0	0.04527	0.176	22.09	5.68	142	228	86.0	0.357	0.533
32.0	593.3	578.6	0.04651	0.163	21.48	6.15	147	226	79.5	0.367	0.529
35.6	619.8	606.1	0.04801	0.149	20.83	6.70	152	224	72.3	0.377	0.523
39.2	649.5	635.1	0.04990	0.135	20.04	7.39	158	222	63.6	0.389	0.520
42.8	678.0	664.3	0.05230	0.121	19.12	8.27	166	218	52.0	0.404	0.508
46.4	710.3	695.6	0.05580	0.103	17.92	9.68	177	211	34.5	0.420	0.494
48.85	731.8	717.1	0.07418	0.074	13.48	13.48	194	194	0	0.460	0.460

* Inches of mercury below one atmosphere.

† Courtesy Linde Air Products Co.

TABLE A.9
PROPERTIES OF DRY SATURATED STEAM*

Temp., F	Abs Press., Lb Sq In.	Specific Volume			Enthalpy			Entropy		
		Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor
<i>t</i>	<i>p</i>	<i>v_f</i>	<i>v_{fg}</i>	<i>v_g</i>	<i>h_f</i>	<i>h_{fg}</i>	<i>h_g</i>	<i>s_f</i>	<i>s_{fg}</i>	<i>s_g</i>
32	0.08854	0.01602	3306	3306	0.00	1075.8	1075.8	0.0000	2.1877	2.1877
35	0.09995	0.01602	2947	2947	3.02	1074.1	1077.1	0.0061	2.1709	2.1770
40	0.12170	0.01602	2444	2444	8.05	1071.3	1079.3	0.0162	2.1435	2.1597
45	0.14752	0.01602	2036.4	2036.4	13.06	1068.4	1081.5	0.0262	2.1167	2.1429
50	0.17811	0.01603	1703.2	1703.2	18.07	1065.6	1083.7	0.0361	2.0903	2.1264
60	0.2563	0.01604	1206.6	1206.7	28.06	1059.9	1088.0	0.0555	2.0393	2.0948
70	0.3631	0.01606	867.8	867.9	38.04	1054.3	1092.1	0.0745	1.9902	2.0647
80	0.5069	0.01608	633.1	633.1	48.02	1048.6	1096.6	0.0932	1.9428	2.0360
90	0.6982	0.01610	468.0	468.0	57.99	1042.9	1100.9	0.1115	1.8972	2.0087
100	0.9492	0.01613	350.3	350.4	67.97	1037.2	1105.2	0.1295	1.8531	1.9826
110	1.2748	0.01617	265.3	265.4	77.94	1031.6	1109.5	0.1471	1.8106	1.9577
120	1.6924	0.01620	203.25	203.27	87.92	1025.8	1113.7	0.1645	1.7694	1.9339
130	2.2225	0.01625	157.32	157.34	97.90	1020.0	1117.9	0.1816	1.7296	1.9112
140	2.8896	0.01629	122.99	123.01	107.89	1014.1	1122.0	0.1984	1.6910	1.8894
150	3.718	0.01634	97.06	97.07	117.89	1008.2	1126.1	0.2149	1.6537	1.8685
160	4.741	0.01639	77.27	77.29	127.89	1002.3	1130.2	0.2311	1.6174	1.8485
170	5.992	0.01645	62.04	62.06	137.90	996.3	1134.2	0.2472	1.5822	1.8293
180	7.510	0.01651	50.21	50.23	147.92	990.2	1138.1	0.2630	1.5480	1.8109
190	9.339	0.01657	40.94	40.96	157.95	984.1	1142.0	0.2785	1.5147	1.7932
200	11.526	0.01663	33.62	33.64	167.99	977.9	1145.9	0.2938	1.4824	1.7762
210	14.123	0.01670	27.80	27.82	178.05	971.6	1149.7	0.3090	1.4508	1.7598
220	17.186	0.01677	23.13	23.15	188.13	965.2	1153.4	0.3239	1.4201	1.7440
230	20.780	0.01684	19.365	19.382	198.23	958.8	1157.0	0.3387	1.3901	1.7288
240	24.969	0.01692	16.306	16.323	208.34	952.2	1160.5	0.3531	1.3609	1.7140
250	29.825	0.01700	13.804	13.821	216.48	945.5	1164.0	0.3675	1.3323	1.6998
260	35.429	0.01709	11.746	11.763	228.64	938.7	1167.3	0.3817	1.3043	1.6860
270	41.858	0.01717	10.044	10.061	238.84	931.8	1170.6	0.3958	1.2769	1.6727
280	49.203	0.01726	8.628	8.645	249.06	924.7	1173.8	0.4096	1.2501	1.6597
290	57.556	0.01735	7.444	7.461	259.31	917.5	1176.8	0.4234	1.2238	1.6472
300	67.013	0.01745	6.449	6.466	269.59	910.1	1179.7	0.4369	1.1980	1.6350
310	77.68	0.01755	5.609	5.626	279.92	902.6	1182.5	0.4504	1.1727	1.6231
320	89.66	0.01765	4.896	4.914	290.28	894.9	1185.2	0.4637	1.1478	1.6115
330	103.06	0.01776	4.289	4.307	300.68	887.0	1187.7	0.4769	1.1233	1.6002
340	118.01	0.01787	3.770	3.788	311.13	879.0	1190.1	0.4900	1.0992	1.5891
350	134.63	0.01799	3.324	3.342	321.63	870.7	1192.3	0.5029	1.0754	1.5783
360	153.04	0.01811	2.939	2.957	332.18	862.2	1194.4	0.5158	1.0519	1.5677
370	173.37	0.01823	2.606	2.625	342.79	853.5	1196.3	0.5286	1.0287	1.5573
380	195.77	0.01836	2.317	2.335	353.45	844.6	1198.1	0.5413	1.0059	1.5471
390	220.37	0.01850	2.0651	2.0836	364.17	835.4	1199.6	0.5539	0.9832	1.5371
400	247.31	0.01864	1.8447	1.8633	374.97	826.0	1201.0	0.5664	0.9608	1.5272
410	276.75	0.01878	1.6512	1.6700	385.83	816.3	1202.1	0.5788	0.9386	1.5174
420	308.83	0.01894	1.4811	1.5000	396.77	806.3	1203.1	0.5912	0.9166	1.5078
430	343.72	0.01910	1.3308	1.3499	407.79	796.0	1203.8	0.6035	0.8947	1.4982
440	381.59	0.01926	1.1979	1.2171	418.90	785.4	1204.3	0.6158	0.8730	1.4887
450	422.6	0.0194	1.0799	1.0996	430.1	774.5	1204.6	0.6280	0.8513	1.4793
460	466.9	0.0196	0.9748	0.9944	441.4	763.2	1204.6	0.6402	0.8298	1.4700
470	514.7	0.0198	0.8811	0.9009	452.8	751.5	1204.3	0.6523	0.8083	1.4606
480	566.1	0.0200	0.7972	0.8172	464.4	739.4	1203.7	0.6645	0.7868	1.4513
490	621.4	0.0202	0.7221	0.7423	476.0	726.8	1202.8	0.6766	0.7653	1.4419
500	680.8	0.0204	0.6545	0.6749	487.8	713.9	1201.7	0.6887	0.7438	1.4325
520	812.4	0.0209	0.5385	0.5594	511.9	686.4	1198.2	0.7130	0.7006	1.4136
540	962.5	0.0215	0.4434	0.4649	536.6	656.6	1193.2	0.7374	0.6568	1.3942
560	1133.1	0.0221	0.3647	0.3868	562.2	624.2	1186.4	0.7621	0.6121	1.3742
580	1325.8	0.0228	0.2989	0.3217	588.9	588.4	1177.3	0.7872	0.5659	1.3532
600	1542.9	0.0236	0.2432	0.2668	617.0	548.5	1165.5	0.8131	0.5176	1.3307
620	1786.6	0.0247	0.1955	0.2201	646.7	503.6	1150.3	0.8398	0.4664	1.3062
640	2059.7	0.0260	0.1538	0.1798	678.6	452.0	1130.5	0.8679	0.4110	1.2789
660	2365.4	0.0278	0.1165	0.1442	714.2	390.2	1104.4	0.8987	0.3485	1.2472
680	2708.1	0.0305	0.0810	0.1115	757.3	309.9	1067.2	0.9351	0.2719	1.2071
700	3093.7	0.0369	0.0392	0.0761	823.3	172.1	995.4	0.9905	0.1484	1.1389
705.4	3206.2	0.0503	0	0.0503	902.7	0	902.7	1.0580	0	1.0580

* Abstracted, by permission, from *Thermodynamic Properties of Steam*, by J. H. Keenan and F. G. Keyes, published by John Wiley & Sons, Inc.

TABLE A.10
PROPERTIES OF SUPERHEATED STEAM *

Temperature—Degrees Fahrenheit													
Abs Press., Lb Sq In. (Sat. Temp.)	200	300	400	500	600	700	800	900	1000	1100	1200	1400	1600
1 (101.74)	392.6 1150.4 2.0512	452.3 1195.8 2.1153	512.0 1241.7 2.1720	571.6 1288.3 2.2233	631.2 1335.7 2.2702	690.8 1383.8 2.3137	750.4 1432.8 2.3542	809.9 1482.7 2.3923	869.5 1533.5 2.4283	929.1 1585.2 2.4625	988.7 1637.7 2.4952	1107.8 1745.7 2.5566	1227.0 1857.5 2.6137
5 (162.24)	78.16 1148.8 1.8718	90.25 1195.0 1.9370	102.26 1241.2 1.9942	114.22 1288.0 2.0456	126.16 1335.4 2.0927	138.10 1383.6 2.1361	150.03 1432.7 2.1767	161.95 1482.6 2.2148	173.87 1533.4 2.2509	185.79 1585.1 2.2851	197.71 1637.7 2.3178	221.6 1745.7 2.3792	245.4 1857.4 2.4363
10 (193.21)	38.85 1146.6 1.7927	45.00 1193.9 1.8595	51.04 1240.6 1.9172	57.05 1287.5 1.9689	63.03 1335.1 2.0160	69.01 1383.4 2.0666	74.98 1432.5 2.1002	80.95 1482.4 2.1383	86.92 1533.2 2.1744	92.88 1585.0 2.2086	98.84 1637.6 2.2413	110.77 1745.6 2.3028	122.69 1857.3 2.3598
14.696 (212.00)	30.53 1192.8 1.8160	34.68 1239.9 1.8743	38.78 1287.1 1.9261	42.86 1334.8 1.9734	46.94 1383.2 2.0170	51.00 1432.3 2.0576	55.07 1482.3 2.0958	59.13 1533.1 2.1319	63.19 1584.8 2.1662	67.25 1637.5 2.1989	75.37 1745.5 2.2603	83.48 1857.3 2.3174
20 (227.96)	22.36 1191.6 1.7808	25.43 1239.2 1.8396	28.46 1286.6 1.8918	31.47 1334.4 1.9392	34.47 1382.9 1.9829	37.46 1432.1 2.0235	40.45 1482.1 2.0618	43.44 1533.0 2.0978	46.42 1584.7 2.1321	49.41 1637.4 2.1648	55.37 1745.4 2.2263	61.34 1857.2 2.2834
40 (267.25)	11.040 1186.8 1.6994	12.628 1236.5 1.7608	14.168 1284.8 1.8140	15.688 1333.1 1.8619	17.198 1381.9 1.9058	18.702 1431.3 1.9467	20.20 1481.4 1.9850	21.70 1532.4 2.0212	23.20 1584.3 2.0555	24.69 1637.0 2.0883	27.68 1745.1 2.1498	30.66 1857.0 2.2069
60 (292.71)	7.259 1181.6 1.6492	8.357 1233.6 1.7135	9.403 1283.0 1.7678	10.427 1331.8 1.8162	11.441 1380.9 1.8605	12.449 1430.5 1.9015	13.452 1480.8 1.9400	14.454 1531.9 1.9762	15.453 1583.8 2.0106	16.451 1636.6 2.0434	18.446 1744.8 2.1049	20.44 1856.7 2.1621
80 (312.03)	6.220 1230.7 1.6791	7.020 1281.1 1.7346	7.797 1330.5 1.7836	8.562 1379.9 1.8281	9.322 1429.7 1.8694	10.077 1480.1 1.9079	10.830 1531.3 1.9442	11.582 1583.4 1.9787	12.332 1636.2 2.0115	13.830 1744.5 2.0731	15.325 1856.5 2.1303
100 (327.81)	4.937 1227.6 1.6518	5.589 1279.1 1.7085	6.218 1329.1 1.7581	6.835 1378.9 1.8029	7.446 1428.9 1.8443	8.052 1479.5 1.8829	8.656 1530.8 1.9193	9.259 1582.9 1.9538	9.860 1635.7 1.9867	11.080 1744.2 2.0484	12.258 1856.2 2.1056
120 (341.25)	4.081 1224.4 1.6287	4.636 1277.2 1.6869	5.165 1327.7 1.7370	5.683 1377.8 1.7822	6.195 1428.1 1.8237	6.702 1478.8 1.8625	7.207 1530.2 1.8990	7.710 1582.4 1.9335	8.212 1635.3 1.9664	9.214 1743.9 2.0281	10.213 1856.0 2.0864
140 (353.02)	3.468 1221.1 1.6087	3.954 1276.2 1.6683	4.413 1326.4 1.7190	4.861 1376.8 1.7645	5.301 1427.3 1.8063	5.738 1478.2 1.8451	6.172 1529.7 1.8817	6.604 1581.9 1.9163	7.035 1634.9 1.9493	7.895 1743.5 2.0110	8.752 1855.7 2.0683

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160 (363.53)	ρ μ κ	3.008 1217.6 1.5908	3.443 1273.1 1.6519	3.849 1325.0 1.7033	4.244 1375.7 1.7491	4.631 1426.4 1.7911	5.015 1477.5 1.8301	5.396 1529.1 1.8667	5.775 1581.4 1.9014	6.152 1634.5 1.9344	6.906 1743.2 1.9962	7.658 1855.5 2.0535
180 (373.06)	ρ μ κ	2.649 1214.0 1.5745	3.044 1271.0 1.6373	3.411 1323.5 1.6894	3.764 1374.7 1.7355	4.110 1425.6 1.7776	4.452 1476.8 1.8167	4.792 1528.6 1.8534	5.129 1581.0 1.8882	5.466 1634.1 1.9212	6.136 1742.9 1.9831	6.804 1855.2 2.0404
200 (381.79)	ρ μ κ	2.361 1210.3 1.5594	2.726 1268.9 1.6240	3.060 1322.1 1.6767	3.380 1373.6 1.7232	3.693 1424.8 1.7655	4.002 1476.2 1.8048	4.309 1528.0 1.8415	4.613 1580.5 1.8763	4.917 1633.7 1.9094	5.521 1742.6 1.9713	6.123 1855.0 2.0287
220 (389.86)	ρ μ κ	2.125 1206.5 1.5453	2.465 1266.7 1.6117	2.772 1320.7 1.6652	3.066 1372.6 1.7120	3.352 1424.0 1.7545	3.634 1475.5 1.7939	3.913 1527.5 1.8308	4.191 1580.0 1.8656	4.467 1633.3 1.8987	5.017 1742.3 1.9607	5.565 1854.7 2.0181
240 (397.37)	ρ μ κ	1.9276 1202.5 1.5319	2.247 1264.5 1.6003	2.533 1319.2 1.6546	2.804 1371.5 1.7017	3.068 1423.2 1.7444	3.327 1474.8 1.7839	3.584 1526.9 1.8209	3.839 1579.6 1.8558	4.093 1632.9 1.8889	4.597 1742.0 1.9510	5.100 1854.5 2.0084
260 (404.42)	ρ μ κ	2.063 1262.3 1.5897	2.330 1317.7 1.6447	2.582 1370.4 1.6922	2.827 1422.3 1.7352	3.067 1474.2 1.7748	3.305 1526.3 1.8118	3.541 1579.1 1.8467	3.776 1632.5 1.8799	4.242 1741.7 1.9420	4.707 1854.2 1.9995
280 (411.05)	ρ μ κ	1.9047 1260.0 1.5796	2.156 1316.2 1.6354	2.392 1369.4 1.6834	2.621 1421.5 1.7265	2.845 1473.5 1.7662	3.066 1525.8 1.8033	3.286 1578.6 1.8393	3.504 1632.1 1.8716	3.938 1741.4 1.9337	4.370 1854.0 1.9912
300 (417.33)	ρ μ κ	1.7675 1257.6 1.5701	2.005 1314.7 1.6268	2.227 1368.3 1.6751	2.442 1420.6 1.7184	2.652 1472.8 1.7582	2.859 1525.2 1.7954	3.065 1578.1 1.8305	3.269 1631.7 1.8638	3.674 1741.0 1.9260	4.078 1853.7 1.9835
350 (431.72)	ρ μ κ	1.4923 1251.5 1.5481	1.7036 1310.9 1.6070	1.8980 1365.5 1.6563	2.084 1418.5 1.7002	2.266 1471.1 1.7403	2.445 1523.8 1.7777	2.622 1577.0 1.8130	2.798 1630.7 1.8463	3.147 1740.3 1.9086	3.493 1853.1 1.9663
400 (444.59)	ρ μ κ	1.2851 1245.1 1.5281	1.4770 1306.9 1.5894	1.6508 1362.7 1.6398	1.8161 1416.4 1.6842	1.9767 1469.4 1.7247	2.134 1522.4 1.7633	2.290 1575.8 1.7977	2.445 1629.6 1.8311	2.751 1739.5 1.8936	3.055 1852.5 1.9513

* Abstracted, by permission, from *Thermodynamic Properties of Steam*, by J. H. Keenan and F. G. Keyes, published by John Wiley & Sons, Inc.

TABLE A.11
SODIUM CHLORIDE SOLUTIONS*
Enthalpy (Btu per lb from 32 F, where it is 200 minus heat of solution)

Sp. Gr.	1.04	1.06	1.08	1.10	1.12	1.14	1.16	1.18
Lb NaCl per 100 lb solution	5.5	8.2	10.9	13.5	16.1	18.6	21.1	23.5
Freezing point, F	26.0	22.7	19.0	14.9	10.4	5.4	-0.3	-3.6
Lb NaCl per gal solution, 60 F	0.48	0.73	0.98	1.24	1.50	1.77	2.04	2.31
Lb solution per gal, 60 F	8.67	8.84	9.00	9.17	9.34	9.50	9.67	9.84
F -6.03	Solid	46.46	50.34	54.35	58.29	62.17	65.96	69.68
-6.03		70.26	86.12	101.77	117.16	132.29	147.11	161.65
-4		72.42	88.88	105.12	121.09	136.79	152.17	167.26
-2		74.65	91.74	108.60	125.19	141.48	157.45	173.13
0	Liquid and Solid	77.05	94.86	112.43	129.71	146.70	163.34	178.98
2		79.66	98.28	116.67	134.74	152.49	169.90	180.56
4		82.50	102.06	121.35	140.33	158.98	177.25	182.29
6		85.61	106.22	126.57	146.57	166.23	183.47	183.74
8		89.09	110.95	132.50	153.73	174.56	185.09	185.33
10		93.10	116.45	139.48	162.16	184.41	186.71	186.92
12		97.72	122.90	147.70	172.13	188.00	188.33	188.51
14		103.14	130.47	157.44	183.96	189.66	189.96	190.11
16		109.60	139.62	169.24	190.85	191.31	191.58	191.71
18		117.62	151.09	184.13	192.54	192.97	193.21	193.30
20		127.97	166.04	193.64	194.24	194.63	194.83	194.90
22		141.96	186.43	195.39	195.94	196.29	196.46	196.50
24		162.72	196.38	197.14	197.63	197.94	198.09	198.10
26		196.19	198.18	198.88	199.33	199.61	199.72	199.70
28		199.00	199.98	200.63	201.03	201.27	201.34	201.30
30		200.85	201.77	202.38	202.74	202.94	202.98	202.91
32	Liquid	202.71	203.57	204.12	204.44	204.60	204.61	204.51
34		204.57	205.37	205.87	206.14	206.26	206.24	206.11
36		206.43	207.17	207.62	207.85	207.93	207.88	207.72
38		208.29	208.97	209.37	209.56	209.60	209.51	209.33
40		210.15	210.78	211.12	211.26	211.27	211.15	210.94
42		212.01	212.57	212.87	212.97	212.94	212.79	212.54
44		213.87	214.38	214.62	214.68	214.61	214.43	214.16
46		215.73	216.18	216.38	216.39	216.29	216.07	215.77
48		217.59	217.99	218.14	218.11	217.96	217.71	217.38
50		219.46	219.80	219.89	219.82	219.63	219.34	218.99

* From *Refrigerating Data Book*, 5th ed. American Society of Refrigerating Engineers, 1943, p. 106.

TABLE A.12
CALCIUM CHLORIDE SOLUTIONS*
Enthalpy (Btu per lb from 32 F, where it is 200 minus heat of solution)

Sp. Gr.	1.08	1.12	1.16	1.20	1.25	1.28
Lb CaCl ₂ per 100 lb solution	9.2	13.5	17.6	21.5	25.1	28.7
Freezing point, F	23.2	16.5	7.0	-5.8	-21.5	-44.3
Lb solution per gal, 60 F	9.01	9.35	9.68	10.01	10.35	10.68
<i>t</i> , F						
-65	Solid	41.14	54.70	67.69	80.22	103.80
-60		43.07	56.59	69.54	82.01	105.52
-59.8		43.15	56.65	69.61	82.08	105.58
-59.8	Liquid and Solid	60.60	82.30	103.13	123.16	142.43
-55		63.34	85.40	106.57	126.16	146.52
-50		66.25	88.67	110.23	130.94	150.87
-45		69.31	92.20	114.18	135.30	155.64
-40		72.57	95.96	118.44	140.04	160.82
-35		76.01	100.02	123.08	145.25	166.58
-30		79.71	104.43	128.15	150.96	172.90
-25		83.72	109.24	133.77	157.34	180.01
-20		88.07	114.63	140.13	164.64	187.57
-15		93.01	120.76	147.41	173.04	190.87
-10		98.64	127.91	156.04	183.06	194.19
-5		105.07	136.36	166.42	192.45	197.51
0		112.60	146.29	178.67	195.95	200.84
5		121.63	158.44	193.87	199.45	204.18
10		133.07	174.13	198.38	202.97	207.53
15		149.39	197.04	202.11	206.49	210.89
20		176.58	201.63	205.86	210.03	214.27
25	Liquid	201.49	205.63	209.62	213.58	217.75
30		205.79	209.65	213.38	217.14	221.04
35		210.12	213.67	217.15	220.70	224.44
40		214.44	217.70	220.94	224.28	227.85
45		218.78	221.74	224.74	227.87	231.27
50		223.11	225.79	228.54	231.47	234.69
55		227.46	229.85	232.35	235.08	238.13
60		231.82	233.91	236.17	238.70	241.58
65		236.18	237.98	240.00	242.32	245.03
70		240.55	242.07	243.83	245.95	248.50
75		244.92	246.17	247.67	249.59	251.97

* From *Refrigerating Data Book*, 5th ed. American Society of Refrigerating Engineers, 1943, p. 102.

TABLE A.13
 VISCOSITY OF FREON REFRIGERANTS, CENTIPOISES*
 (Liquid at saturation pressure; vapor at 1 atm.)†

Temp. F	F-11 CCl ₃ F		F-12 CCl ₂ F ₂		F-22 CHClF ₂	
	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor
-40	.980	.0088	.423	.0106	.351	.0105
-20	.801	.0092	.371	.0110	.316	.0109
0	.677	.0095	.335	.0113	.291	.0113
20	.586	.0099	.300	.0116	.271	.0118
40	.517	.0103	.286	.0120	.256	.0122
60	.401	.0106	.269	.0123	.243	.0126
80	.417	.0110	.255	.0126	.232	.0130
100	.380	.0113	.242	.0120	.223	.0133
120	.349	.0116	.232	.0130	.214	.0137
140	.323	.0120	.222	.0135	.207	.0141
160	.300	.0123	.214	.0138	.201	.0145
180	.281	.0126	.207	.0140	.195	.0148

* From "The Viscosities of 'Freon' Refrigerants," by A. F. Benning and W. H. Markwood, *Refrigerating Engineering*, Vol. 37, No. 4 (April, 1939), p. 243.

† Effect of pressure changes of less than one atmosphere is negligible.

TABLE A.14
CONDUCTIVITY OF INSULATING MATERIALS*
Btu(in.) per (hr)(sq ft)(F)

Material	Density lb per cu ft	Mean temp., F	Conduc- tivity <i>k</i>
Asbestos, packed.....	43.8	32	1.62
Asbestos, loose.....	29.3	32	1.07
Asbestos, paper, thin layers organic binder.....	31.2	86	0.49
Asbestos mill board.....	60.5	86	0.84
Asbestos wood.....	123.0	86	2.70
Asphalt roofing (felt).....	55.0	86	0.70
Balsa.....	7.36	86	0.35
Cement wood (sawdust and Portland Cement).....	44.6	68	0.97
Charcoal (hardwoods), coarse.....	13.2	90	0.36
Cork board, typical.....	8.3	35	0.27
Cork board, typical.....	8.3	60	0.28
Cork board.....	6.7	-22	0.22
Cork, regranulated, coarse.....	8.1	90	0.31
Cork, regranulated.....	6.5	77	0.27
Cotton.....	5.06	32	0.39
Diatomaceous earth.....	30.0	30	0.56
Eel grass.....	9.4	86	0.31
Glass wool, high grade.....	1.76	85	0.265
Glass wool, high grade.....	3.61	74	0.237
Glass wool, high grade.....	5.37	75	0.224
Glass wool, commercial grade.....	2.49	65	0.26
Hair felt, not compressed.....	13.0	90	0.26
Insulation boards, fiber various.....	15-21	70	0.32-0.38
Kapok, loosely packed.....	0.87	84	0.24
Planer shavings, various woods.....	8.7	86	0.40
Rock wool.....	10.0	90	0.27
Rock wool.....	14.0	90	0.28
Rubber, expanded.....	4.85	72	0.21
Sawdust, various woods.....	12.0	90	0.41
Miscellaneous materials:			
Glass, window.....	5.5
Gravel.....	115.0	68	2.6
Gravel, peanut, 2.6 % water.....	68	7.5
Ice.....	57.5	32	15.6
Soil, clay, 14 % moisture			
Loosely packed.....	75	68	2.6
Loaded 1 cwt. per sq. ft.....	80	68	4.9
Loaded 1 ton per sq. ft.....	96	68	8.4
Loam over sand and gravel, 3 ft. deep. Seasonal change due to variation of moisture content.....	7.8-10.7
Snow.....	34.7	..	3.2
Water.....	50	4.3

* From *Refrigerating Data Book*, 5th ed. American Society of Refrigerating Engineers, 1943, p. 155.

TABLE A.15
THERMAL CONDUCTIVITY OF BUILDING MATERIALS*
 Btu(in.) per (hr)(sq ft)(F)

Material	Density lb per cu ft	Mean temp., F	Conduc- tivity <i>k</i>
Brick, low density	5.0
Brick, high density	9.2
Cement mortar	12.0
Cement plaster, typical	12.0
Concrete, typical	12.0
Concrete, typical fiber gypsum, 87.5 % gypsum, 12.5 % wood chips	51.2	74	1.66
Sand and gravel	142	75	12.6
Limestone	132	75	10.8
Cinder	97	75	4.9
Cinders, boiler ($\frac{1}{2}$ to $\frac{3}{4}$ in.)	60	71	1.23
Stucco, typical	12.0
Tile or terazzo, typical	12.0
Asbestos building board	123	86	2.7
Gypsum, between layers of heavy paper	62.8	70	1.41
Gypsum, plaster, typical	3.30
Wood across grain, typical	1.00
Balsa	8.8	90	0.38
Redwood, California, 16 % moisture	22	75	0.74
Fir, Douglas, 16 % moisture	26	75	0.76
Maple, hard, 16 % moisture	40	75	1.15
Pine, yellow short leaf, 16 % moisture	26	75	0.84
Pine, yellow short leaf, 16 % moisture	36	75	1.04

* From *Refrigerating Data Book*, 5th ed. American Society of Refrigerating Engineers, 1943, p. 156.

TABLE A.16
CONDUCTANCE OF BUILDING MATERIALS AND CONSTRUCTIONS*
 Btu per (hr)(sq ft)(F) for thickness indicated

Material	Density lb per cu ft	Mean temp., F	Conduct- ance C
Plasterboard, $\frac{3}{8}$ in. thick.....	3.73
Plasterboard, $\frac{1}{2}$ in. thick.....	2.82
Roofs:			
Asphalt, composition or prepared roofing.....	70	75	6.50
Shingles, asbestos.....	65	75	6.00
Shingles, asphalt.....	70	75	6.50
Shingles, slate.....	201	..	10.4
Metal lath and plaster, total thickness $\frac{3}{4}$ in.....	4.40
Wood lath and plaster, total thickness $\frac{3}{4}$ in.....	...	70	2.50
Fir sheathing, 1-in. and building paper.....	...	30	0.86
Fir sheathing, 1-in. building paper and yellow pine lap siding.....	...	20	0.50
Fir sheathing, 1-in. building paper and stucco.....	...	20	0.82
Air spaces over $\frac{3}{4}$ in. faced ordinary building materials.....	...	40	1.10
Typical hollow clay tile, 4 in.....	1.00
Typical hollow clay tile, 6 in.....	0.64
Typical hollow clay tile, 8 in.....	0.60
Typical hollow clay tile, 10 in.....	0.58
Typical hollow clay tile, 12 in.....	0.40
Typical hollow clay tile, 16 in.....	0.31
Concrete block, sand and gravel, typical, 8 in.....	1.00
Concrete block, sand and gravel, typical, 12 in.....	0.80
Concrete block, cinder aggregate, typical, 8 in.....	0.60
Concrete block, cinder aggregate, typical, 12 in.....	0.53
Surface coefficient, inside f_i	1.65
Surface coefficient, outside f_o , 15 m.p.h. wind.....	6.00

* From *Refrigerating Data Book*, 5th ed. American Society of Refrigerating Engineers, 1943, p. 156.

TABLE A.17
THERMODYNAMIC PROPERTIES OF WATER AT SATURATION*

Fahr. Temp. (F)	Absolute Pressure In. Hg	Specific Volume, cu ft per lb		Enthalpy, Btu per lb	
		Sat. Solid	Sat. Vapor	Sat. Solid	Sat. Vapor
<i>t</i>	<i>p_s</i>	<i>v_i</i>	<i>v_g</i>	<i>h_i</i>	<i>h_g</i>
-160	1.008/10 ⁷	0.01722	36.07 · 10 ³	-222.05	990.38
-155	1.840	0.01723	20.08	-220.44	992.58
-150	3.29	0.01723	11.39	-218.82	994.80
-145	5.803	0.01724	6.577	-217.17	997.00
-140	10.03	0.01724	3.864	-215.49	999.21
-135	17.06	0.01725	2.308	-213.80	1001.42
-130	28.56	0.01725	1.400	-212.08	1003.63
-125	4.708/10 ⁶	0.01726	8.622 · 10 ⁷	-210.34	1005.84
-120	7.649	0.01726	5.386	-208.58	1008.05
-115	12.26	0.01727	3.411	-206.79	1010.26
-110	19.38	0.01728	2.189	-204.98	1012.47
-105	30.25	0.01728	1.422	-203.14	1014.68
-100	46.64	0.01729	0.9352	-201.28	1016.89
-95	7.108/10 ⁵	0.01729	6.223 · 10 ⁶	-199.40	1019.10
-90	10.71	0.01730	4.186	-197.49	1021.31
-85	15.96	0.01730	2.846	-195.56	1023.52
-80	23.55	0.01731	1.955	-193.60	1025.73
-75	34.39	0.01732	1.356	-191.62	1027.94
-70	49.74	0.01732	0.9501	-189.61	1030.15
-65	71.28	0.01733	0.6715	-187.58	1032.36
-60	1.012/10 ⁵	0.01734	4.788 · 10 ⁵	-185.52	1034.58
-55	1.426	0.01734	3.443	-183.44	1036.79
-50	1.900	0.01735	2.496	-181.34	1039.00
-45	2.757	0.01736	1.824	-179.21	1041.21
-40	3.790	0.01737	1.343	-177.06	1043.42
-35	5.170	0.01737	0.9961	-174.88	1045.63
-30	0.7003/10 ⁴	0.01738	7.441 · 10 ⁴	-172.68	1047.84
-25	0.9420	0.01739	5.596	-170.46	1050.05
-20	1.259	0.01739	4.237	-168.21	1052.26
-15	1.670	0.01740	3.228	-165.94	1054.47
-10	2.203	0.01741	2.475	-163.65	1056.67
-5	2.888	0.01741	1.909	-161.33	1058.88
0	0.03764/1	0.01742	14.81 · 10 ³	-158.98	1061.09
5	0.04878	0.01743	11.55	-156.61	1063.29
10	0.06286	0.01744	9.060	-154.22	1065.50

* Abstracted, by permission, from *Heating Ventilating Air Conditioning Guide* 1948, Chapter 3.

TABLE A.17
THERMODYNAMIC PROPERTIES OF WATER AT SATURATION (CONT.)

Fahr. Temp. (F)	Absolute Pressure In. Hg	Specific Volume, cu ft per lb		Enthalpy, Btu per lb	
		Sat. Solid or Liquid	Sat. Vapor	Sat. Solid or Liquid	Sat. Vapor
<i>t</i>	<i>p_s</i>	<i>v_i</i>	<i>v_g</i>	<i>h_i</i>	<i>h_g</i>
15	0.08056	0.01744	7.144	-151.80	1067.70
20	0.1027	0.01745	5.662	-149.36	1069.90
25	0.1303	0.01746	4.509	-146.89	1072.09
30	0.1645	0.01747	3.608	-144.40	1074.29
32	0.1803	0.01747	3.305	-143.40	1075.16
		<i>v_f</i>		<i>h_f</i>	
32	0.18036	0.01602	3304.6 · 1	0.00	1075.16
35	0.20342	0.01602	2947.8	3.02	1076.48
40	0.24767	0.01602	2445.4	8.04	1078.68
45	0.30023	0.01602	2037.3	13.06	1080.87
50	0.36240	0.01602	1704.3	18.07	1083.06
55	0.43564	0.01603	1431.5	23.08	1085.24
60	0.52160	0.01603	1207.1	28.08	1087.42
65	0.62209	0.01604	1021.7	33.08	1089.60
70	0.73916	0.01605	867.97	38.07	1091.78
75	0.87506	0.01606	739.97	43.06	1093.95
80	1.0323	0.01607	633.03	48.05	1096.12
85	1.2136	0.01609	543.35	53.05	1098.28
90	1.4219	0.01610	467.90	58.04	1100.44
95	1.6607	0.01612	404.19	63.03	1102.59
100	1.9334	0.01613	350.22	68.02	1104.74

TABLE A.18

THERMODYNAMIC PROPERTIES OF MOIST AIR (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG)*

Fahr. Temp. (F)	Humidity Ratio	Volume cu ft/lb dry air		Enthalpy Btu/lb dry air		Condensed Water	
		Dry Air	Moist Air	Dry Air	Moist Air	Enthalpy Btu/lb	Entropy Btu/(lb) (F)
<i>t</i>	$W_a \times 10^3$	v_a	v_s	h_a	h_s	h_w	s_w
-160	0.2120	7.520	7.520	-38.504	-38.504	-222.00	-0.4907
-150	0.6932	7.775	7.775	-36.088	-36.088	-218.77	-0.4800
-140	2.109	8.029	8.029	-33.674	-33.674	-215.44	-0.4695
-130	6.000	8.283	8.283	-31.262	-31.262	-212.03	-0.4590
	$W_a \times 10^7$						
-120	1.606	8.537	8.537	-28.852	-28.852	-208.52	-0.4485
-110	4.063	8.792	8.792	-26.444	-26.444	-204.92	-0.4381
-100	9.772	9.046	9.046	-24.037	-24.036	-201.23	-0.4277
	$W_a \times 10^6$						
-90	2.242	9.300	9.300	-21.631	-21.629	-197.44	-0.4173
-80	4.930	9.553	9.553	-19.225	-19.220	-193.55	-0.4069
-70	10.40	9.806	9.806	-16.820	-16.809	-189.56	-0.3965
	$W_a \times 10^5$						
-60	2.118	10.059	10.059	-14.416	-14.394	-185.47	-0.3861
-50	4.163	10.313	10.314	-12.012	-11.969	-181.29	-0.3758
-40	7.925	10.566	10.567	-9.609	-9.526	-177.01	-0.3655
	$W_a \times 10^4$						
-30	1.464	10.820	10.822	-7.207	-7.053	-172.64	-0.3552
-20	2.630	11.073	11.078	-4.804	-4.527	-168.17	-0.3449
-10	4.606	11.326	11.334	-2.402	-1.915	-163.60	-0.3346
	$W_a \times 10^3$						
0	0.7872	11.578	11.593	0.000	0.835	-158.93	-0.3244
5	1.020	11.705	11.724	1.201	2.286	-156.57	-0.3193
10	1.315	11.831	11.856	2.402	3.803	-154.17	-0.3141
15	1.687	11.958	11.990	3.603	5.403	-151.76	-0.3090
20	2.152	12.084	12.126	4.804	7.106	-149.31	-0.3039
25	2.733	12.211	12.265	6.005	8.934	-146.85	-0.2988
30	3.454	12.338	12.406	7.206	10.915	-144.36	-0.2936
35	4.275	12.464	12.549	8.407	13.008	3.06	0.00061
40	5.213	12.590	12.695	9.608	15.230	8.09	0.0162
45	6.331	12.717	12.846	10.809	17.650	13.10	0.0262
50	7.658	12.843	13.001	12.010	20.301	18.11	0.0361
51	7.952	12.868	13.032	12.250	20.862	19.11	0.0381
52	8.256	12.894	13.064	12.491	21.436	20.11	0.0400
53	8.569	12.919	13.097	12.731	22.020	21.12	0.0420
54	8.894	12.944	13.129	12.971	22.615	22.12	0.0439
55	9.229	12.970	13.162	13.211	23.22	23.12	0.0459
56	9.575	12.995	13.195	13.452	23.84	24.12	0.0478
57	9.934	13.020	13.228	13.692	24.48	25.12	0.0497
58	10.30	13.045	13.261	13.932	25.12	26.12	0.0517
59	10.69	13.071	13.295	14.172	25.78	27.12	0.0536

* Abstracted, by permission, from *Heating Ventilating Air Conditioning Guide* 1948, Chapter 3.

TABLE A.18
THERMODYNAMIC PROPERTIES OF MOIST AIR (STANDARD ATMOSPHERIC PRESSURE,
29.921 IN. HG) (CONT.)

Fahr. Temp. (F)	Humidity Ratio	Volume cu ft/lb dry air		Enthalpy Btu/lb dry air		Condensed Water	
		Dry Air	Moist Air	Dry Air	Moist Air	Enthalpy Btu/lb	Entropy Btu/(lb) (F)
<i>t</i>	$W_a \times 10^3$	v_a	v_a	h_a	h_a	h_w	s_w
60	11.08	13.096	13.329	14.413	26.46	28.12	0.0555
65	13.26	13.222	13.504	15.614	30.06	33.11	0.0651
	$W_a \times 10^3$						
70	1.582	13.348	13.687	16.816	34.09	38.11	0.0746
75	1.882	13.474	13.881	18.018	38.61	43.10	0.0840
76	1.948	13.499	13.921	18.259	39.57	44.10	0.0859
77	2.016	13.525	13.962	18.499	40.57	45.10	0.0877
78	2.086	13.550	14.003	18.740	41.58	46.10	0.0896
79	2.158	13.575	14.045	18.980	42.62	47.10	0.0914
80	2.233	13.601	14.087	19.221	43.69	48.10	0.0933
81	2.310	13.626	14.130	19.461	44.78	49.09	0.0952
82	2.389	13.651	14.174	19.702	45.90	50.09	0.0970
83	2.471	13.676	14.218	19.942	47.04	51.09	0.0989
84	2.555	13.702	14.262	20.183	48.22	52.09	0.1007
85	2.642	13.727	14.308	20.423	49.43	53.09	0.1025
86	2.731	13.752	14.354	20.663	50.66	54.08	0.1043
87	2.824	13.777	14.401	20.904	51.93	55.08	0.1062
88	2.919	13.803	14.448	21.144	53.23	56.08	0.1080
89	3.017	13.828	14.496	21.385	54.56	57.08	0.1098
90	3.118	13.853	14.545	21.625	55.93	58.08	0.1116
91	3.223	13.879	14.595	21.865	57.33	59.07	0.1135
92	3.330	13.904	14.645	22.106	58.78	60.07	0.1153
93	3.441	13.929	14.697	22.346	60.25	61.07	0.1171
94	3.556	13.954	14.749	22.587	61.77	62.07	0.1188
95	3.673	13.980	14.802	22.827	63.32	63.07	0.1206
96	3.795	14.005	14.856	23.068	64.92	64.06	0.1224
97	3.920	14.030	14.911	23.308	66.55	65.06	0.1242
98	4.049	14.056	14.967	23.548	68.23	66.06	0.1260
99	4.182	14.081	15.023	23.789	69.96	67.06	0.1278
100	4.319	14.106	15.081	24.029	71.73	68.06	0.1296
101	4.460	14.131	15.140	24.270	73.55	69.05	0.1314
102	4.606	14.157	15.200	24.510	75.42	70.05	0.1332
103	4.756	14.182	15.261	24.751	77.34	71.05	0.1350
104	4.911	14.207	15.324	24.991	79.31	72.05	0.1367

APPENDIX

TABLE A.19

PROPERTIES OF MIXTURES OF AIR AND SATURATED WATER VAPOR, 0 TO 32 F*
Total Pressure—29.921 in. Hg (Vapor Pressures Are Those of Sub-cooled Water)

Tem- pera- ture, F	Satu- ration pressure, in. Hg	Specific humidity for satu- ration, grains per lb of dry air	Volume of satu- rated mixture, cu ft per lb of dry air	Specific enthalpy of vapor- ization of water, Btu per lb	Specific enthalpy of satu- rated steam, Btu per lb	Specific enthalpy of dry air, Btu per lb	Enthalpy of mixture, Btu per lb of dry air	Σ Function of mixture, Btu per lb of dry air	Tem- pera- ture, F
0	0.0448	6.53	11.61	1093.8	1061.8	0	0.99	1.02	0
1	0.0469	6.84	11.63	1093.2	1062.2	0.24	1.28	1.31	1
2	0.0492	7.17	11.66	1092.7	1062.7	0.48	1.57	1.60	2
3	0.0515	7.51	11.68	1092.1	1063.1	0.72	1.86	1.89	3
4	0.0540	7.87	11.71	1091.5	1063.5	0.96	2.16	2.19	4
5	0.0565	8.24	11.74	1091.0	1064.0	1.20	2.45	2.48	5
6	0.0591	8.63	11.77	1090.4	1064.0	1.44	2.75	2.78	6
7	0.0619	9.03	11.79	1089.9	1064.9	1.68	3.05	3.09	7
8	0.0647	9.45	11.82	1089.3	1065.3	1.92	3.36	3.39	8
9	0.0677	9.89	11.84	1088.7	1065.7	2.16	3.67	3.70	9
10	0.0708	10.33	11.87	1088.2	1066.2	2.40	3.97	4.01	10
11	0.0741	10.81	11.89	1087.6	1066.6	2.64	4.29	4.32	11
12	0.0775	11.31	11.92	1087.1	1067.1	2.88	4.60	4.64	12
13	0.0810	11.82	11.95	1086.5	1067.5	3.12	4.92	4.95	13
14	0.0846	12.35	11.97	1085.9	1067.9	3.36	5.24	5.28	14
15	0.0884	12.90	12.00	1085.4	1068.4	3.60	5.57	5.60	15
16	0.0923	13.47	12.02	1084.8	1068.8	3.84	5.90	5.93	16
17	0.0964	14.08	12.04	1084.3	1069.3	4.08	6.23	6.26	17
18	0.1007	14.71	12.07	1083.7	1069.7	4.32	6.57	6.60	18
19	0.1051	15.36	12.10	1083.1	1070.1	4.56	6.91	6.94	19
20	0.1097	16.02	12.13	1082.6	1070.6	4.80	7.25	7.28	20
21	0.1145	16.73	12.16	1082.0	1071.0	5.04	7.60	7.63	21
22	0.1194	17.45	12.19	1081.5	1071.5	5.28	7.95	7.98	22
23	0.1245	18.20	12.21	1080.9	1071.9	5.52	8.31	8.33	23
24	0.1298	18.97	12.24	1080.3	1072.3	5.76	8.67	8.69	24
25	0.1353	19.78	12.27	1079.8	1072.8	6.00	9.03	9.05	25
26	0.1411	20.63	12.30	1079.2	1073.2	6.24	9.40	9.42	26
27	0.1470	21.51	12.33	1078.7	1073.7	6.48	9.78	9.79	27
28	0.1532	22.41	12.35	1078.1	1074.1	6.72	10.16	10.17	28
29	0.1597	23.36	12.38	1077.5	1074.5	6.96	10.94	10.56	29
30	0.1663	24.33	12.41	1076.9	1074.9	7.20	10.94	10.94	30
31	0.1732	25.35	12.44	1076.3	1075.3	7.44	11.33	11.34	31
32	0.1803	26.41	12.46	1075.8	1075.8	7.68	11.74	11.74	32

* From *Refrigerating Data Book*, 5th ed. American Society of Refrigerating Engineers, 1943, p. 434.

TABLE A.20
PROPERTIES OF DRY AIR AT 14.7 PSIA
(Units: Btu, lb, hr, ft, F)

Temp.	Specific Heat	Conductivity	Viscosity	Density
0	0.240	0.0124	0.040	0.0863
20	0.240	0.0128	0.041	0.0827
40	0.240	0.0132	0.042	0.0794
60	0.240	0.0136	0.043	0.0763
80	0.240	0.0140	0.045	0.0734
100	0.240	0.0145	0.046	0.0708

TABLE A.21
DIMENSIONS OF WELDED AND SEAMLESS STEEL PIPE*

Nominal pipe size, in.	Outside diameter, in.	Nominal Wall Thickness for Schedule Numbers, in.	
		40	80
$\frac{1}{8}$	0.405	0.068	0.095
$\frac{1}{4}$	0.540	0.088	0.119
$\frac{3}{8}$	0.675	0.091	0.126
$\frac{1}{2}$	0.840	0.109	0.147
$\frac{3}{4}$	1.050	0.113	0.154
1	1.315	0.133	0.179
$1\frac{1}{4}$	1.660	0.140	0.191
$1\frac{1}{2}$	1.900	0.145	0.200
2	2.375	0.154	0.218
$2\frac{1}{2}$	2.875	0.203	0.276
3	3.5	0.216	0.300
$3\frac{1}{2}$	4.0	0.226	0.318
4	4.5	0.237	0.337
5	5.563	0.258	0.375
6	6.625	0.280	0.432
8	8.625	0.322	0.500
10	10.75	0.365	0.593
12	12.75	0.406	0.687

* From American Standard for Welded and Seamless Steel Pipe (ASA No. B36-10, 1939); copies available from American Standards Association, 70 E. 45th St., New York.

TABLE A.22
EQUIVALENT FEET OF PIPE FOR VALVES AND FITTINGS*

(Based on data presented in Crane Company's Technical Paper No. 409, May, 1942. Values shown are average values based on standard-weight pipe and Type L tubing.)

Line Size, Inches	IPS	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	5	6	8	10	12
	OD	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{7}{8}$	$1\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{5}{8}$	$2\frac{1}{8}$	$2\frac{5}{8}$	$3\frac{1}{8}$	$3\frac{5}{8}$	$4\frac{1}{8}$	$5\frac{1}{8}$	$6\frac{1}{8}$	$8\frac{1}{8}$	$10\frac{1}{8}$	$12\frac{1}{8}$
Globe valve (open) . .		14	16	22	28	36	42	57	69	83	99	118	138	168	225	280	335
Angle valve (open) . .		7	9	12	15	18	21	28	34	42	49	57	70	83	117	140	165
Standard elbow		1	2	2	3	4	4	5	7	8	10	12	14	16	20	26	31
Standard tee (through side outlet)		3	4	5	6	8	9	12	14	17	20	22	28	34	44	56	65

* Reproduced, by permission, from *Equipment Standards*, 1946 edition of the Air Conditioning and Refrigerating Machinery Association, Incorporated.

TABLE A.23
MAXIMUM TONS REFRIGERATION FOR AMMONIA MAINS*

Pipe size in.	Suction line			Discharge line	Liquid Line	
	Suction pressure psig (temp F)				Condenser to receiver	Receiver to system
	5 (−17.2 F)	20 (5.5 F)	45 (30 F)			
$\frac{3}{8}$	2.5	12.0
$\frac{1}{2}$	0.6	1.1	2.0	3.1	6.0	20.0
$\frac{3}{4}$	1.2	2.2	4.1	6.0	14.0	75.0
1	2.2	4.0	7.5	11.4	24.0	137.
1 $\frac{1}{4}$	4.4	8.0	15.0	22.4	50.0	245.
1 $\frac{1}{2}$	6.4	11.8	21.6	30.9	77.0	400.
2	12.1	22.2	42.0	62.0	140.	850.
2 $\frac{1}{2}$	19.1	35.5	65.0	97.5	220.	1475.
3	31.5	59.0	108.	160.	375.	2400.
3 $\frac{1}{2}$	46.6	87.5	156.	238.	540.	3500.
4	64.0	118.	240.	330.	740.	
5	117.	208.	385.	560.	1320.	
6	175.	306.	600.	905.	2030.	
8	362.	650.	1200.	1810.	4200.	
10	640.	1180.	2160.	3200.		
12	940.	1850.				

* Reproduced, by permission, from *Equipment Standards*, 1946 edition of the Air Conditioning and Refrigerating Machinery Association, Incorporated.

TABLE A.24
TONS REFRIGERATION FOR METHYL CHLORIDE MAINS*
FOR 1 PSI PRESSURE DROP PER 100 FT†

Line size in.	Internal cross sectional area, sq in.	Suction line			Discharge line, 86 F cond temp (80.82 psig)	Liquid line
		Suction temp F and corresponding press. psig				
		0 F (4.0 lb)	20 F (14.06 lb)	40 F (27.91 lb)		
$\frac{1}{8}$ OD	0.031					0.46
$\frac{1}{8}$ OD	0.078	0.09	0.12	0.15	0.19	1.55
$\frac{1}{8}$ OD	0.146	0.22	0.27	0.34	0.45	3.62
$\frac{1}{8}$ IPS	0.191	0.27	0.34	0.43	0.57	4.6
$\frac{1}{8}$ OD	0.233	0.42	0.52	0.66	0.87	7.0
$\frac{1}{4}$ IPS	0.304	0.51	0.64	0.80	1.06	8.5
$\frac{1}{4}$ OD	0.484	1.11	1.40	1.77	2.33	18.8
$\frac{3}{8}$ IPS	0.533	1.11	1.40	1.77	2.33	18.8
1 $\frac{1}{8}$ OD	0.825	2.28	2.87	3.62	4.77	38.4
1 IPS	0.861	2.12	2.67	3.37	4.44	35.8
1 $\frac{1}{2}$ OD	1.256	3.89	4.89	6.17	8.12	65.4
1 $\frac{1}{2}$ IPS	1.496	4.28	5.38	6.78	8.92	72.0
1 $\frac{3}{8}$ OD	1.780	6.35	8.0	10.1	13.3	107
1 $\frac{3}{8}$ IPS	2.036	6.6	8.3	10.5	13.8	111
2 $\frac{1}{8}$ OD	3.094	13.2	16.6	21.0	27.6	222
2 IPS	3.356	13.0	16.3	20.6	27.1	218
2 $\frac{1}{2}$ OD	4.77	23.7	29.8	37.6	49.6	399
2 $\frac{1}{2}$ IPS	4.78	21.4	26.9	33.9	44.7	360
3 $\frac{1}{8}$ OD	6.812	37.6	47.2	59.5	78.5	632
3 IPS	7.383	36.2	45.6	57.5	75.9	610
3 $\frac{1}{2}$ OD	9.213	57.0	71.8	90.5	119	960
3 $\frac{1}{2}$ IPS	9.687	54.4	68.5	86.3	114	916
4 $\frac{1}{8}$ OD	11.97	80.2	101	127	168	1350
4 IPS	12.73	78.4	96.2	121	160	1287
5 IPS	19.99	139	174	220	290	2350
6 IPS	28.89	227	286	360	474	
8 IPS	50.03	466	587	740	975	
10 IPS	78.82	867	1093	1380	1820	
12 IPS	113.1	1400	1760	2220	2920	

* Hendrickson, H. M., "Determination of Refrigerant Pipe Size." *Refrigerating Engineering*, Vol. 52, No. 4 (October, 1946), p. 317.

† Factors for other pressure drops per 100 ft

Press. drop	Multiplier
0.5	0.69
2.0	1.45
3.0	1.80
4.0	2.10
5.0	2.36

TABLE A.25
FREON-12 LIQUID LINES, TONS CAPACITY PER 100 FT EQUIVALENT LENGTH*

Line size, inches	Pressure drop per 100 ft equivalent length, psi			
	3	5	10	20
$\frac{3}{8}$ OD	0.88	1.14	1.80	2.58
$\frac{1}{2}$ OD	2.89	3.64	5.56	8.50
$\frac{1}{2}$ IPS	4.86	6.81	10.2	15.8
$\frac{3}{4}$ OD	4.86	6.81	10.2	15.8
$\frac{3}{4}$ IPS	9.73	12.6	18.5	27.0
$\frac{7}{8}$ OD	10.5	14.1	21.8	33.0
1 IPS	21.4	28.2	41.3	60.8
1 $\frac{1}{8}$ OD	21.4	28.2	41.3	60.8
1 $\frac{1}{8}$ IPS	36.9	48.1	70.5	101.
1 $\frac{3}{8}$ OD	36.9	48.1	70.5	101.
1 $\frac{3}{8}$ IPS	62.0	80.2	114.	160.
1 $\frac{5}{8}$ OD	62.0	80.2	114.	160.
2 IPS	124.	161.	231.	328.
2 $\frac{1}{2}$ IPS	230.	297.	426.	607.
3 IPS	364.	469.	676.	972.
3 $\frac{1}{2}$ IPS	539.	704.	1005.	1430.
4 IPS	753.	972.	1385.	1945.

* Reproduced, by permission, from *Equipment Standards*, 1946 edition of the Air Conditioning and Refrigerating Machinery Association, Incorporated.

TABLE A.26
MAXIMUM TONS OF COMPRESSOR CAPACITY FOR FREON-12 LINES*
 (Only for temperatures indicated)†

Line Size Inches	Suction Lines Based on 105 F Condensing Temperature						Discharge Lines	
	Psi Pressure Drop per 100 Ft Equivalent Length at 40 F Saturation						Condensing Temperature	
	$\frac{1}{8}$	1	2	3	4	5	115 F	90 F
$\frac{1}{8}$ OD	0.14	0.20	0.28	0.35	0.41	0.45		
$\frac{1}{8}$ IPS	0.17	0.24	0.34	0.42	0.49	0.54		
$\frac{1}{4}$ OD	0.25	0.35	0.51	0.62	0.73	0.81	1.43	1.15
$\frac{1}{4}$ IPS	0.35	0.45	0.65	0.79	0.93	1.03	1.87	1.50
$\frac{3}{8}$ OD	0.55	0.76	1.10	1.34	1.58	1.75	2.97	2.38
$\frac{3}{8}$ IPS	0.68	0.94	1.35	1.65	1.92	2.12	3.26	2.62
$1\frac{1}{8}$ OD	1.26	1.80	2.57	3.17	3.76	4.15	5.05	4.05
1 IPS	1.43	2.01	2.89	3.54	4.17	4.60	5.29	4.25
$1\frac{1}{2}$ OD	2.21	3.12	4.45	5.50	6.38	7.05	7.72	6.19
$1\frac{1}{2}$ IPS	2.70	3.82	5.37	6.72	7.68	8.48	9.16	7.35
$1\frac{3}{4}$ OD	3.40	4.78	6.79	8.42	9.77	10.8	10.92	8.75
$1\frac{3}{4}$ IPS	4.05	5.75	8.10	10.12	11.6	12.8	12.5	10.0
$2\frac{1}{8}$ OD	6.12	8.60	12.1	15.1	17.4	19.2	19.2	15.3
2 IPS	7.66	10.9	15.3	19.2	22.2	24.5	20.6	16.5
$2\frac{1}{2}$ OD	12.0	17.1	24.0	30.1	34.6	38.2	32.2	25.9
$2\frac{1}{2}$ IPS	12.0	17.1	24.0	30.1	34.6	38.2	32.2	25.9
$3\frac{1}{8}$ OD	19.1	27.2	38.2	47.8	55.0	60.7	51.5	39.8
3 IPS	20.9	29.4	42.3	51.8	60.0	66.2	54.5	43.8
$3\frac{1}{2}$ OD	27.8	39.7	55.7	69.8	80.3	88.7	72.0	57.6
$3\frac{1}{2}$ IPS	30.2	43.2	61.0	76.1	87.0	96.0	78.8	63.3
$4\frac{1}{8}$ OD	38.6	55.2	78.0	97.3	111	123	95.8	77.1
4 IPS	40.7	58.6	83.0	103	118	130	101.6	81.6
5 IPS	71.3	100	141	176	203	224	171.5	137.8
6 IPS	126	183	257	322	366	403	266	214
8 IPS	211	297	422	523	602	664	461	370
10 IPS	352	503	712	887	1024	1130	725	582
12 IPS	550	780	1106	1373	1582	1748	1041	836

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† See Table 12.5 for other suction temperatures, p. 283.

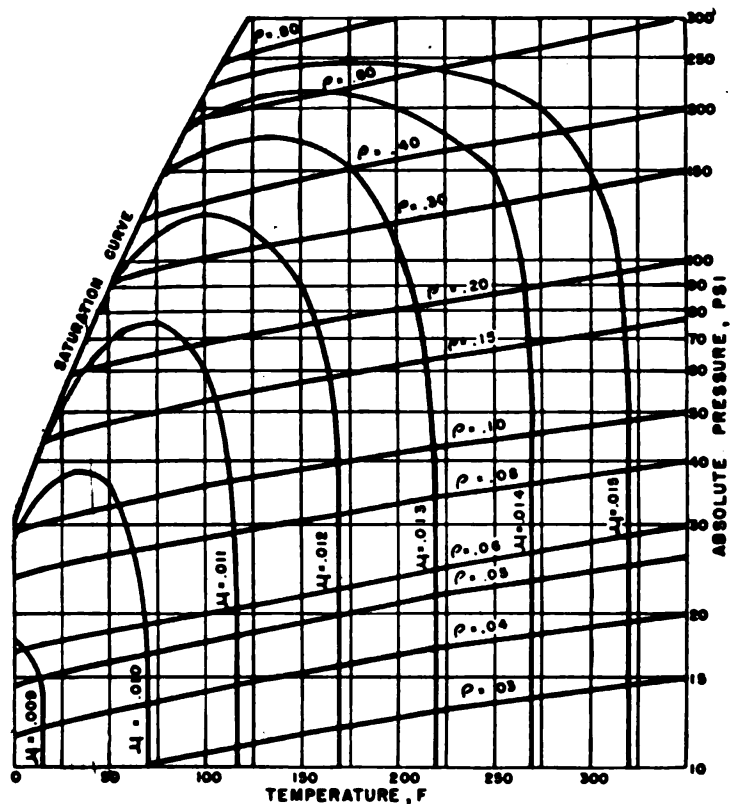


Fig. A.1. Viscosity and density of ammonia vapor.
 From *Refrigerating Data Book*, 5th ed., 1943, p. 178.
 American Society of Refrigerating Engineers.

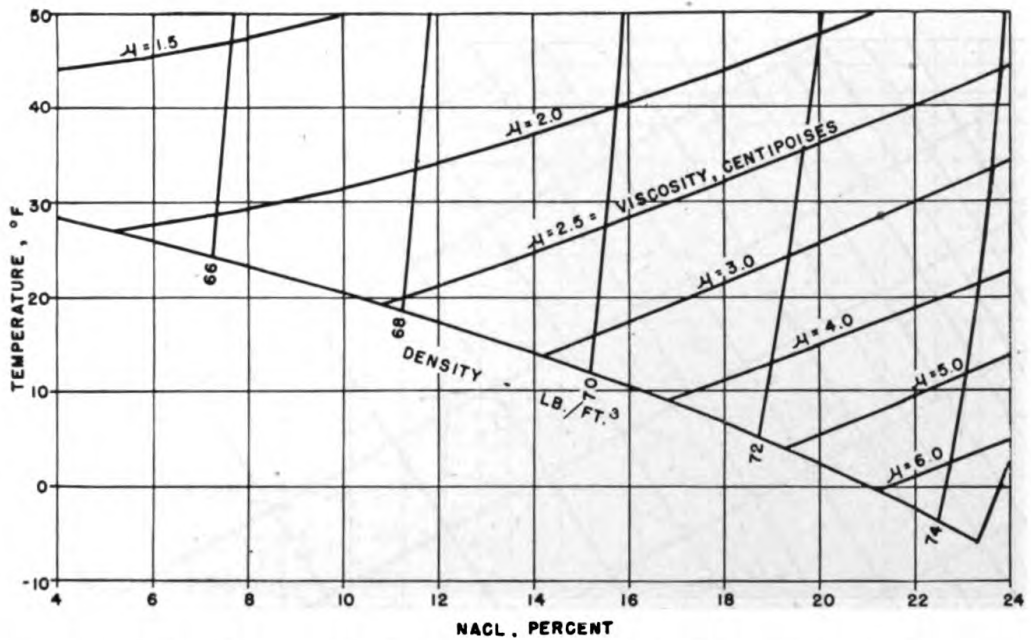


Fig. A.2. Viscosity and density of sodium chloride brine. From *Refrigerating Data Book*, 5th ed., 1943, p. 182. American Society of Refrigerating Engineers.

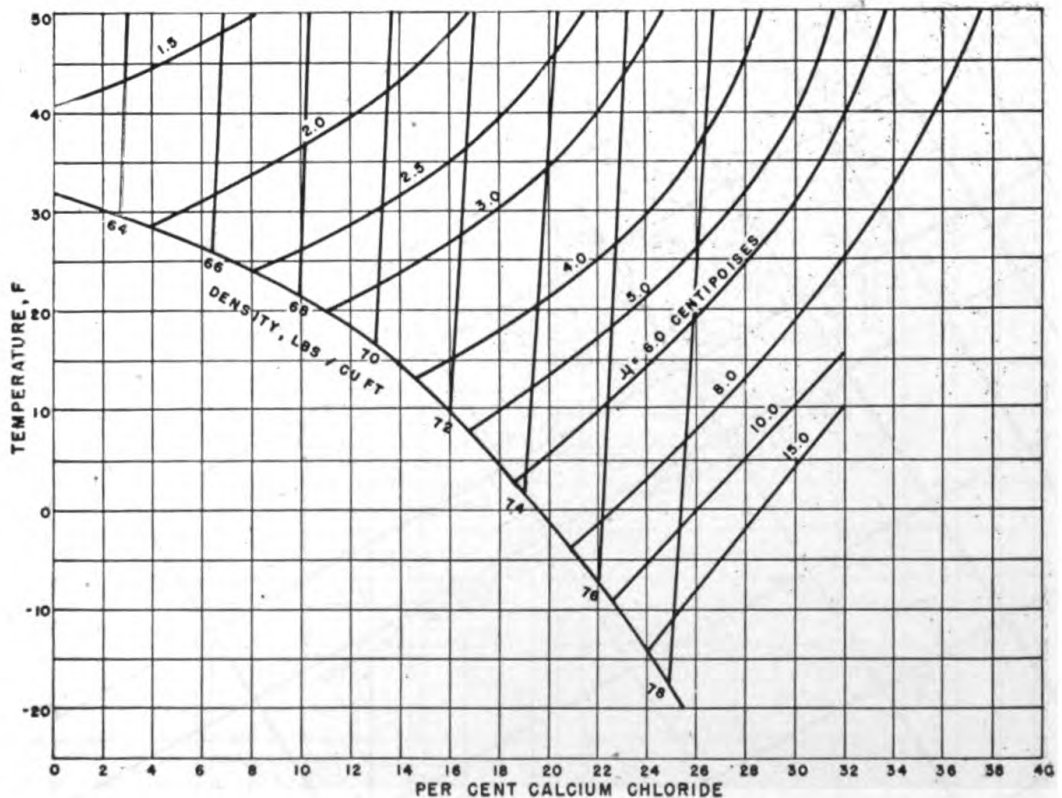


Fig. A.3. Viscosity and density of calcium chloride brine. From *Refrigerating Data Book*, 5th ed., 1943, p. 180. American Society of Refrigerating Engineers.

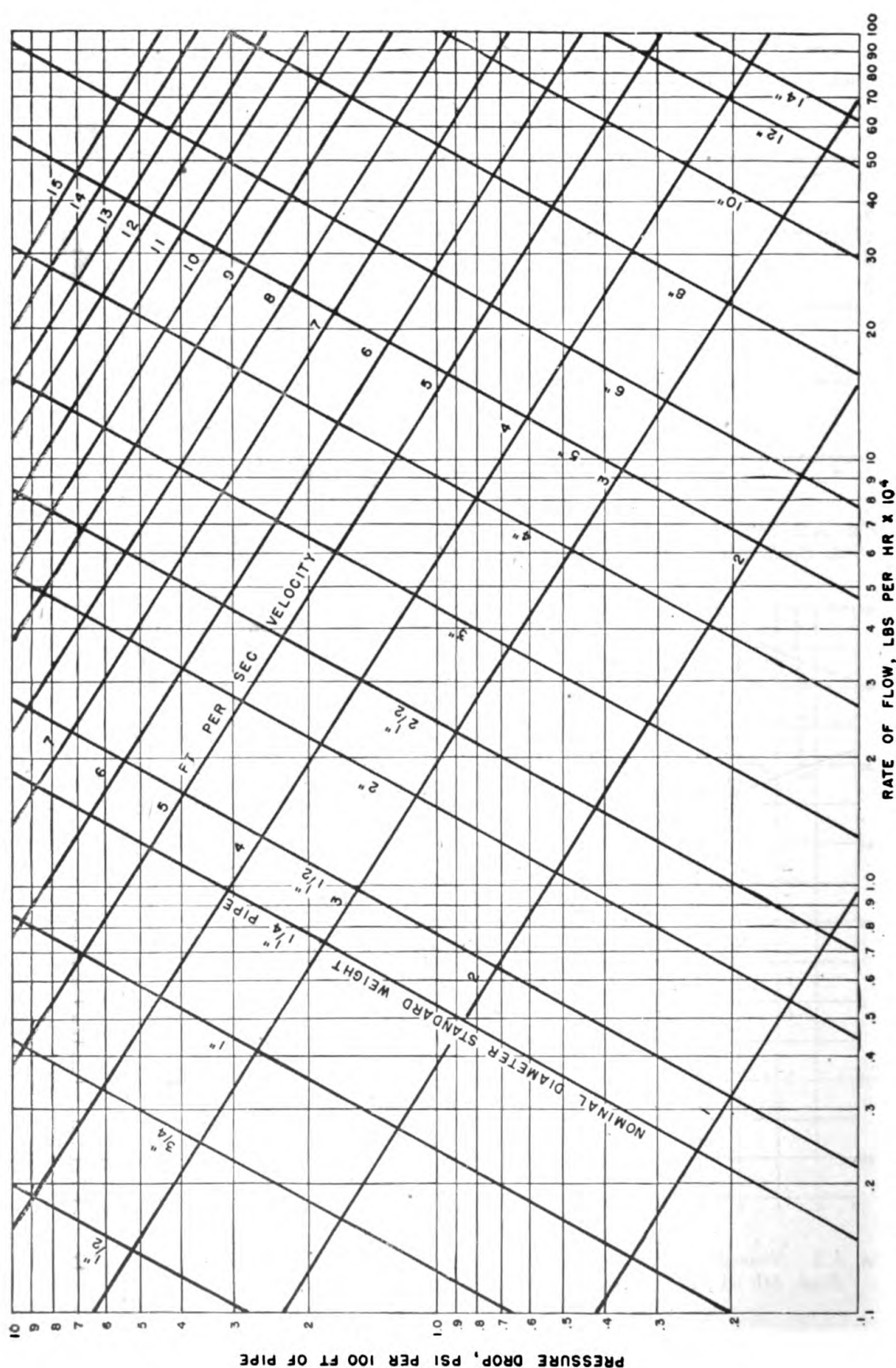


Fig. A.4 Pressure drop for water in pipe of fair condition. From Crocker, Sabin, *Piping Handbook*. New York: McGraw-Hill Book Company, Inc. 1949 - 1950

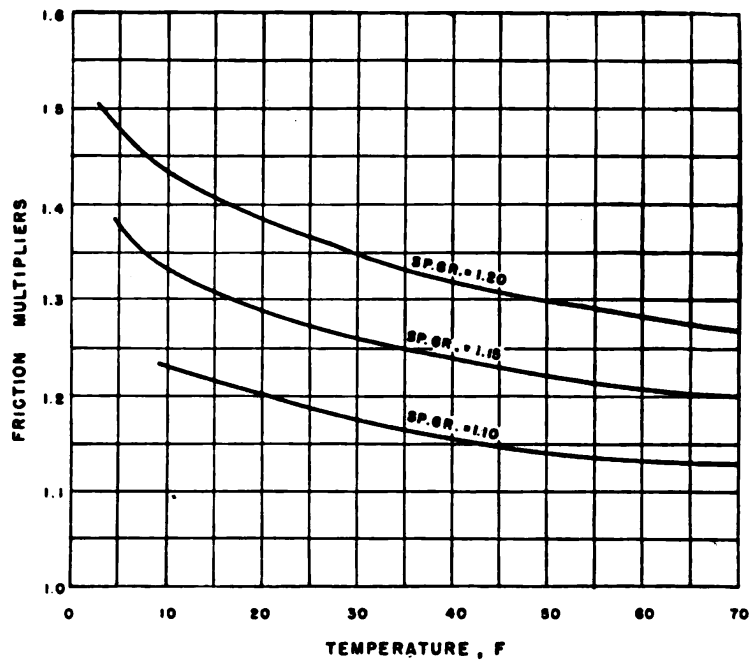


Fig. A.5. Pipe friction multipliers for sodium chloride brine. From R. C. Doremus, "Piping for Indirect Refrigeration." *Heating, Piping and Air Conditioning*, Vol. 10, No. 3 (April, 1938).

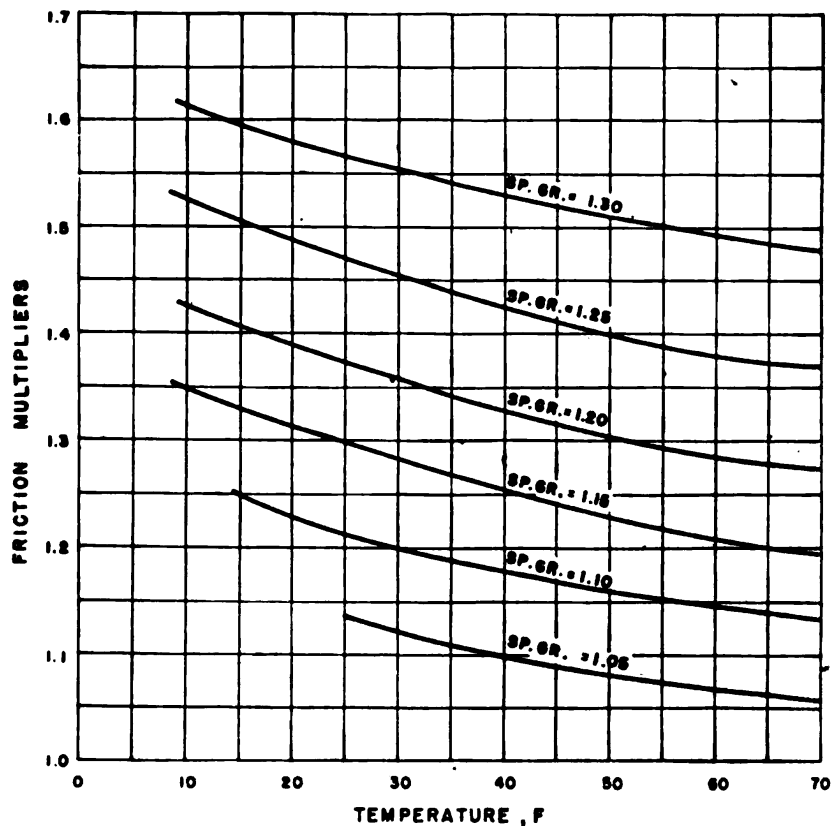
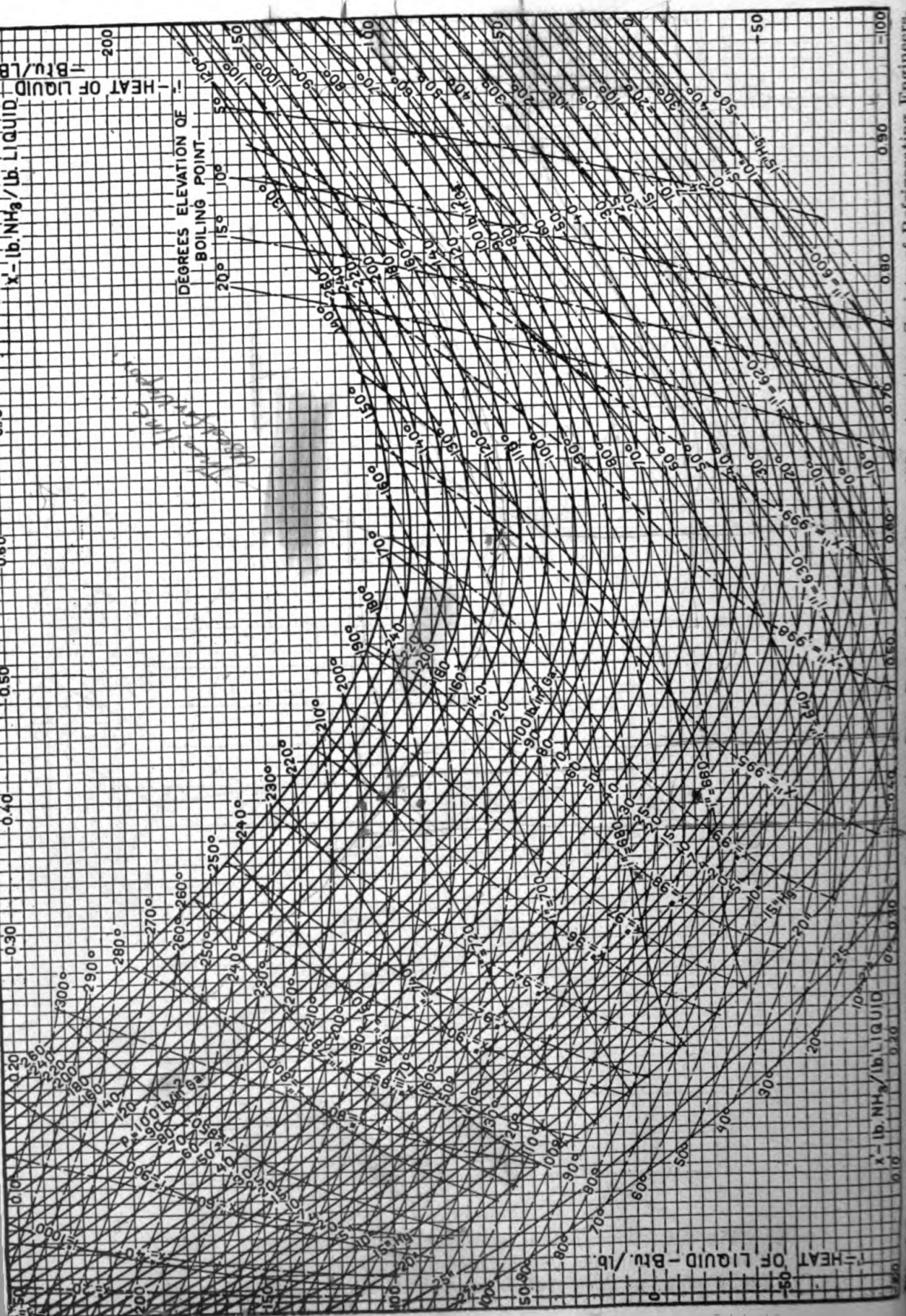


Fig. A.6. Pipe friction multipliers for calcium chloride brine. From R. C. Doremus, "Piping for Indirect Refrigeration." *Heating, Piping and Air Conditioning*, Vol. 10, No. 4 (April, 1938).



Answers to Problems

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2.1. 23.5 tons

2.3. 1.02 tons

2.5. 8.14 tons

CHAPTER 3

3.1. $n = 1.19$

3.3. (a) $x = 0.16$

(b) 1.163 cu ft per lb

(c) $x = 0.14$

3.5. 3.016 Btu per lb

3.7. $\frac{897 \text{ Btu per hr}}{12.3 \text{ Btu per lb}}$

3.9. 12.3 Btu per lb

CHAPTER 4

4.1. (a) c.p. = 14.17; 15.17

(b) c.p. = 8.75; 9.75

(c) c.p. = 6.18; 7.18

(d) c.p. = 4.69; 5.69

4.3. (a) $\eta = 0.982$; 0.952; 0.852

(b) $\eta = 0.963$; 0.902; 0.690

4.5. (a) 148.0 Btu per lb

(b) 1.351 lb per (ton)(min)

(c) 6.72 cu ft per (ton)(min)

(d) 1.1 hp per ton

(e) c.p. = 4.29

(f) 246.6 Btu per (ton)(min)

4.7. (a) 18.6 per cent; 100 per cent

(b) 18.6 per cent; 89.6 per cent

4.8. (a) 6.42 tons

(b) 6.42 hp

(c) 6.74 hp

(d) 1644.2 Btu per hr

(e) 95,761 Btu per hr

4.11. (a) 3.42×5.13 in.

(b) 4.12×6.17 in.

(c) 7.47×11.2 in.

4.13. (a) 10.0 F

(b) c.p. = 6.80 (no heat exchanger)

c.p. = 6.84 (with heat exchanger)

4.14. (a) 72.87 Btu per lb

(b) 54.89 lb per min

(c) 144.85 cu ft per min

(d) 28.2 hp

(e) c.p. = 3.35

(f) 5194.2 Btu per min

(g) 3.29 in.

CHAPTER 5

5.1. (a) 2.91 lb

(b) 0.0145 lb

(c) 0.00277 lb

5.3. 4.22 lb per min

5.5. (a) 8,850 to 9,440 lb

(b) 20 to 30 lb

(c) nonflammable; 17.04 to 27.6 lb

5.6. (a) 1603 lb; 23.3 cu ft

(b) 1203 lb; 17.5 cu ft

5.9. (a) 128.4 lb

(b) 1.74 cu ft

CHAPTER 6

6.1. (a) 141.3 lb per min

(b) 394.2 cu ft per min

(c) 299.2 cu ft per min

(d) c.p. = 2.35

(e) 2.01 hp

6.3. (a) 156.25 F

(b) 242.5 F

(c) 28.3 hp

(d) 122.5 F

(e) -11.9 F

(f) 31.98 hp

(g) 5.21 tons

6.4. (a) 1.362 lb motive steam per lb vapor

(b) $x = 0.937$

(c) 974.1 Btu per lb

(d) 16.8 lb per (hr)(ton)

(e) 22,524 cu ft per (hr)(ton)

- 8.7. (a) 250 tons
(b) 48.6 lb per min; 116,402 cu ft per min
8.9. (a) 2.88 lb per (ton)(min)
(b) 24.5 cu ft per (ton)(min)
(c) 7.38 cu ft per (ton)(min)
- (d) 0.730 hp per ton
(e) c.p. = 6.47
(f) 371.9 gpm
- 6.12. (a) 19.9 psia
(b) 0.00876 hp; 6.53 watts
(c) 340.0 watt-hr

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8.3. (a) 0.48 psi
(b) 1.13 psi
(c) 1.49 psi
8.5. 20.2 Btu per (hr)(sq ft of inside surface)(F)
8.7. 1.93 Btu per (hr)(ft of pipe)
8.9. 0.643 sq ft
- 8.11. (a) 5 F
(b) 75 F
8.13. (a) 0.063 Btu per (hr)(sq ft)(F)
(b) 0.0453 Btu per (hr)(sq ft)(F)
8.15. (a) 0.483 Btu per (hr)(sq ft)(F)
(b) 0.092 Btu per (hr)(sq ft)(F)
8.17. (a) 0.108 Btu per (hr)(sq ft)(F)
(b) 0.104 Btu per (hr)(sq ft)(F)

CHAPTER 9

- 9.1. (a) 0.126 in. Hg
(b) 15.9 per cent
(c) 15.5 per cent
(d) 0.00263 lb
(e) 20.16 Btu per lb
(f) 20.34 Btu per lb
(g) 20.11 Btu per lb
9.3. (a) 30.2 per cent
(b) 30.1 per cent
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(e) 7.54 Btu per lb
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- 9.5. 80 F
68.6 F
9.7. Fog at 53.5 F
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9.13. 41,600 lb
29.5 F
98 per cent
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10.9. 49,020 Btu per hr

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(b) 11.35 lb per min
11.5. 2
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25 F
11.9. 2170 Btu per (hr)(sq ft)(F)

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(b) 1 and $\frac{1}{2}$, 3, and $1\frac{1}{4}$ in.
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- (b) 6.66 psi
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(b) 5.50 cu ft per min
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(b) 1.41 per cent
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(b) 113 Btu
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(d) 601 Btu
(e) -51.4 Btu
(f) 21.6 Btu
(g) 996 Btu
- (h) 803 Btu
(i) 6 Btu
(j) 1131 Btu
(k) 158 Btu
(m) 668 Btu
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(b) refrigeration

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(c) $\mu = 0.056$
 $x = 0.951$
yield = 4.9 per cent
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(b) -98.03 F
(c) 6.1 F
(d) -76.3 F
- 16.4. (a) 10.3; 7.35
(b) 2.28; 3.85 lb per (ton)(min)
(c) 1.27; 1.38 hp per ton
(d) 37.8; 11.72 cu ft per (ton)(min)
(e) c.p. = 3.70; 3.42
combined c.p. = 2.26
(f) 53.3; 66.5 F
(g) 2.65 hp
(h) 49.5 cu ft per (ton)(min)
- (i) compression ratio = 49.4
3.26 lb per (ton)(min)
110.5 cu ft per (ton)(min)
3.40 hp per ton
c.p. = 1.39
discharge temperature = 169.8 F
- 16.6. (a) 853.4 psia
(b) 47.57 lb
(c) 2.43 lb
(d) 8988.1 Btu
- 16.8. (a) 37.4 lb
(b) \$1.31
- 16.9. (a) 186.7 hp per (ton solid CO₂ per hr)
(b) 594.3 cu ft per (ton solid CO₂) (min)

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8.6 to 27.2 tons
\$6,000 to \$15,000
\$4,500 to \$9,000
- 18.5. (a) 2100 sq ft
4600 sq ft
- 4700 sq ft
350 seats
(b) \$7350 +
\$10,350 +
\$8225 +
\$7875 +

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